MODERN MECHANICAL ENGINEERING

A PRACTICAL TREATISE
WRITTEN BY SPECIALISTS

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COAL AND ASH HANDLING PLANT

BY

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Coal and Ash Handling Plant

CHAPTER I

Coal Handling

The question of the economical handling of coal for industrial purposes is undoubtedly of great importance and may be considered from two main points of view; first, the efficient handling and transporting from the colliery to the point at which it is delivered to the consumer; and second, the method to be employed by the fuel user for taking delivery of the coal and either storing it or distributing it to his various fuel-utilizing plants, such as boilers, retorts, gas producers, and other furnaces.

It is not proposed to deal with the question of handling the fuel between the colliery and the point where it is delivered to the consumer, as this subject is outside the scope of the present article, and therefore the object in what follows is to place before the reader a description of some of the various methods and types of apparatus which are used for handling coal economically after it has been delivered to the consumer by road, rail, or water.

In connection with electric power plant, where the size of the generating stations is continually being increased owing to the rapid development of power requirements, there is a field in which the question of coal handling is of paramount importance, as the cost of generation of electrical energy depends largely on low power-station costs, which in the absence of efficient and reliable coal handling plant are impossible.

The following deals chiefly with plant for use in power stations as described above, and in which it will be appreciated that, as continuity of supply is the first consideration, it is essential to install plant which is not only efficient and economical but which shall be immune from periodical breakdowns and involuntary stoppages. In this connection, therefore, it is unwise to cut down prices and purchase inferior plant, as the losses sustained by stoppage and breakdowns rapidly absorb any saving in initial cost.

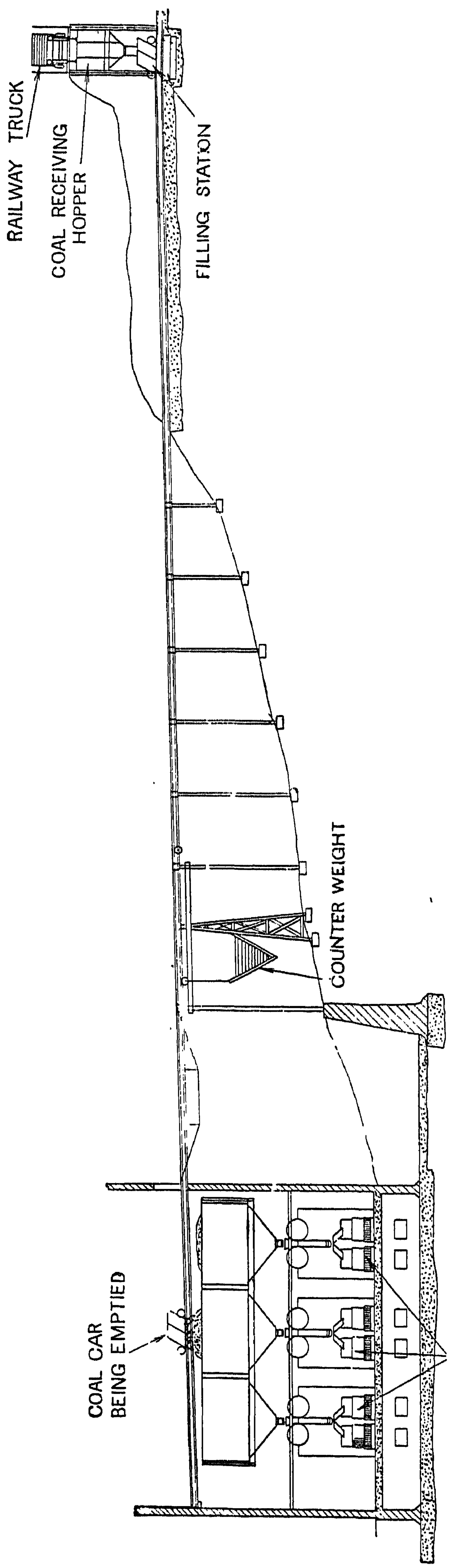

Fig. 1.—Diagrammatic Arrangement of Babcock & Wilcox Automatic Railway

GENERAL METHODS

By Sidings Direct to Coal Bunkers.—In certain cases it will be found that the layout of the power station plant, and the natural formation and level of the ground on which it stands in relation to the adjacent railway sidings, render it possible to provide an elevated railway siding leading to the coal storage bunkers situated over the boilers in the boiler house. This arrangement enables the coal trucks to be shunted directly over the bunkers and emptied therein without the provision of any special coal handling plant. This scheme is probably the cheapest and most convenient arrangement to adopt, as the amount of plant to be maintained in efficient working order is reduced to a minimum. All that is required is an overhead siding, over which a shunting locomotive can handle the coal trucks direct from the railway company's sidings. The overhead siding may also be used for depositing coal on the ground below for storage purposes. The coal, after being emptied out of the trucks on to the storage space, can be stacked by means of a jib crane and grab over a relatively large area on each side of the overhead siding, thereby permitting a considerable amount of coal to be stored in readiness for emergencies, such as a failure of the coal supply by rail due to accident, strikes, &c. The crane which is used to distribute the coal on the storage space can also be used to reclaim the coal and load it into trucks for shunting on to the bunkers over the boilers.

If the general layout of the plant and the arrangement of the site is such that this scheme cannot be adopted, the use of an automatic railway may be considered, as described below.

By Automatic Railway.—In certain cases it is not possible to adopt the scheme of sidings direct to the bunkers referred to above, as, although the contour of the ground may be suitable, the relative positions of the railway company's sidings and the power station coal bunkers may

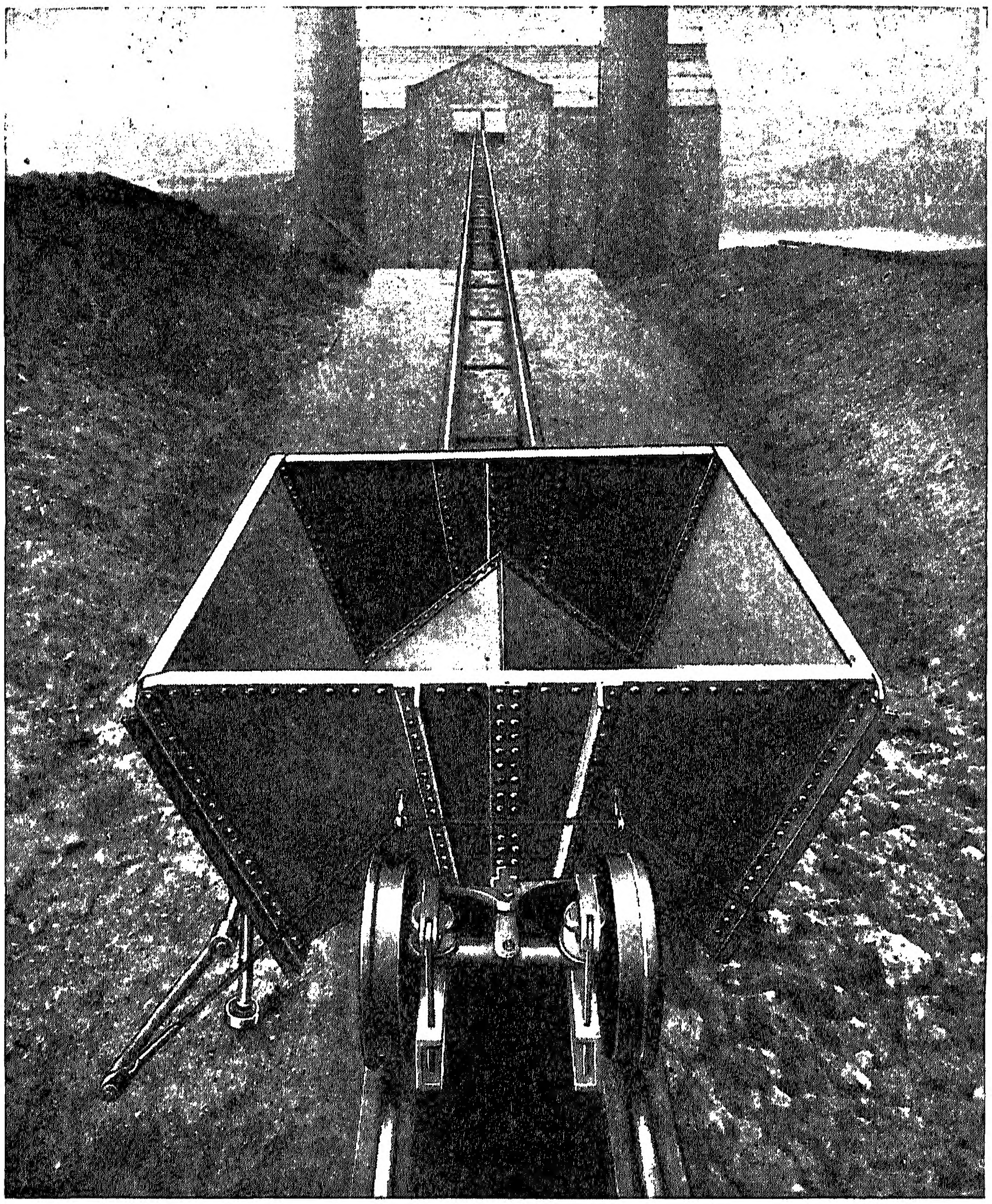

Fig. 2.—Automatic Railway. Capacity of automatic car, 1 ton ≡ 50 tons per hour

be such that sidings cannot be arranged. In these cases it is often found convenient to install an automatic railway, one example of which is shown in figs. 1 and 2. This plant is manufactured by Messrs. Babcock & Wilcox, and is arranged so that the gradient of the railway is sufficient to allow the descending loaded car to supply the power necessary for returning it to the starting-point when empty.

The makers state that a 3 per cent gradient is sufficient for running, dumping, and returning the car to the filling station. The automatic railway may be as long as 600 ft. or more, and the full car is simply started

on the down grade from the filling station by an attendant, while the dumping and returning of the empty car is automatically performed.

The full car descends the gradient against the action of a counterweight, shown in fig. 1, which is connected to the car by a flexible steel cable running over a system of guide pulleys. When the car reaches the bunkers over the boilers it is emptied by the automatic opening of the car sides, and the action of the counterweight then returns the car to the filling station. At the filling station a coal-receiving hopper is preferably provided, from which the coal is filled into the automatic coal car. If desired, the weight of the coal in each car can be weighed and automatically recorded. An automatic railway such as described can make fifty trips per hour, and thus deal with 50 tons per hour, when equipped with a 1-ton car, or 100 tons per hour with a 2-ton car.

The two systems of railways referred to above can, however, only be used in those instances where the arrangement of the plant and the site is suitable.

Generally it is found that the railway siding level coincides with the level of the power station yard, and in these cases it is necessary to use some system of elevators and conveyors to lift the coal from coal-receiving hoppers below ground level into the coal bunkers over the boilers.

The main types of elevators and conveyors so used are as follows:

1. The Gravity Bucket Conveyor for lifting and distributing the coal to the bunkers.

2. The Bucket Chain Elevator for lifting the coal, combined with either a tray or a belt conveyor, for distributing it to the bunkers.

3. Belt Conveyors throughout.

These are described in Chapter II.

CHAPTER II

Coal Handling Apparatus

The Gravity Bucket Conveyor.—This conveyor consists of an endless chain of tipping buckets linked together by chains of special construction. The whole system is balanced so that the work of driving the complete mechanism round is simply the power required to overcome the friction of the loaded chain of buckets, plus the work done in lifting the coal from the point at which it is fed into the buckets to the height required.

A typical arrangement of power station, with a gravity bucket coal elevator, is shown in fig. 3, in which the coal is dropped from railway wagons into a coal-receiving hopper A, whence the small coal passes through a jigging screen B (which also regulates the feed of coal from the hopper), while the larger pieces which pass over the screen are broken to the required size in a roll type breaker C. After passing the screen and the breaker the

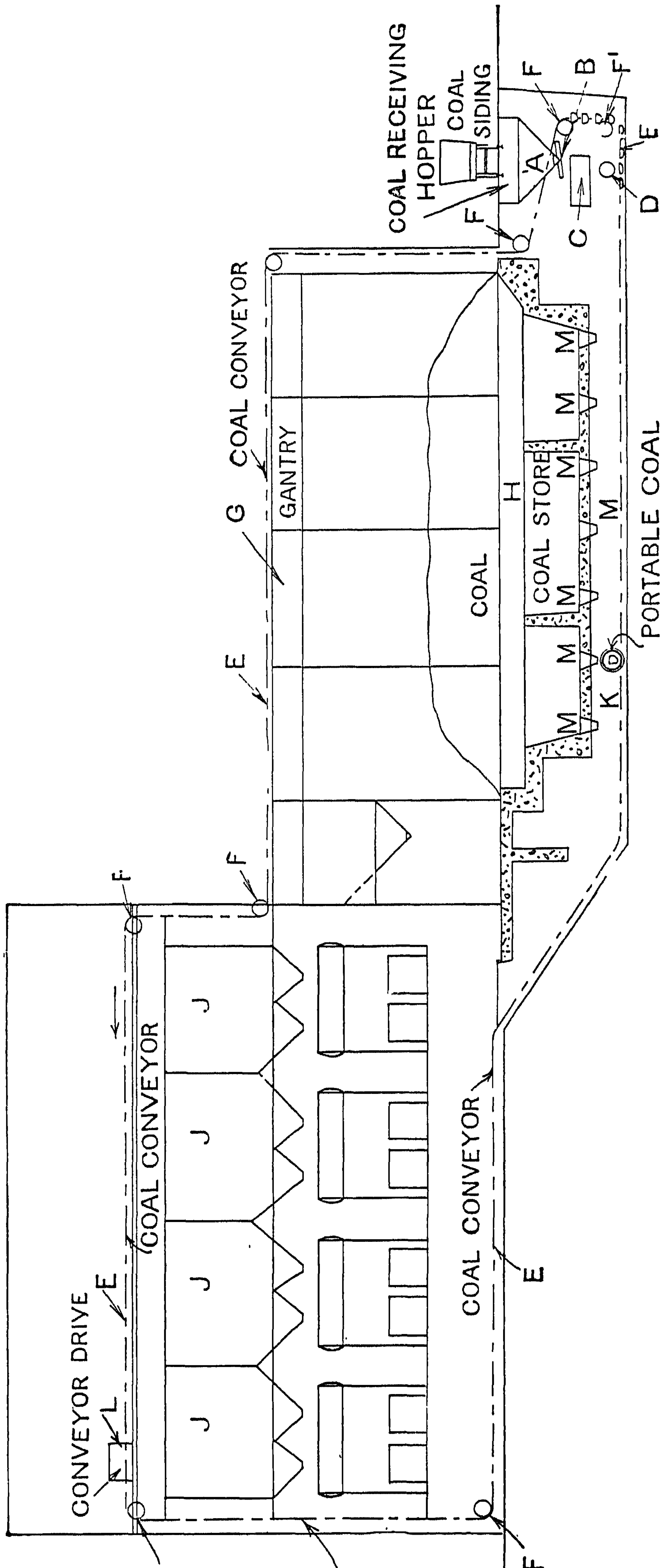

Fig. 3.—Typical Arrangement of Power Station with Gravity Bucket Coal Elevator

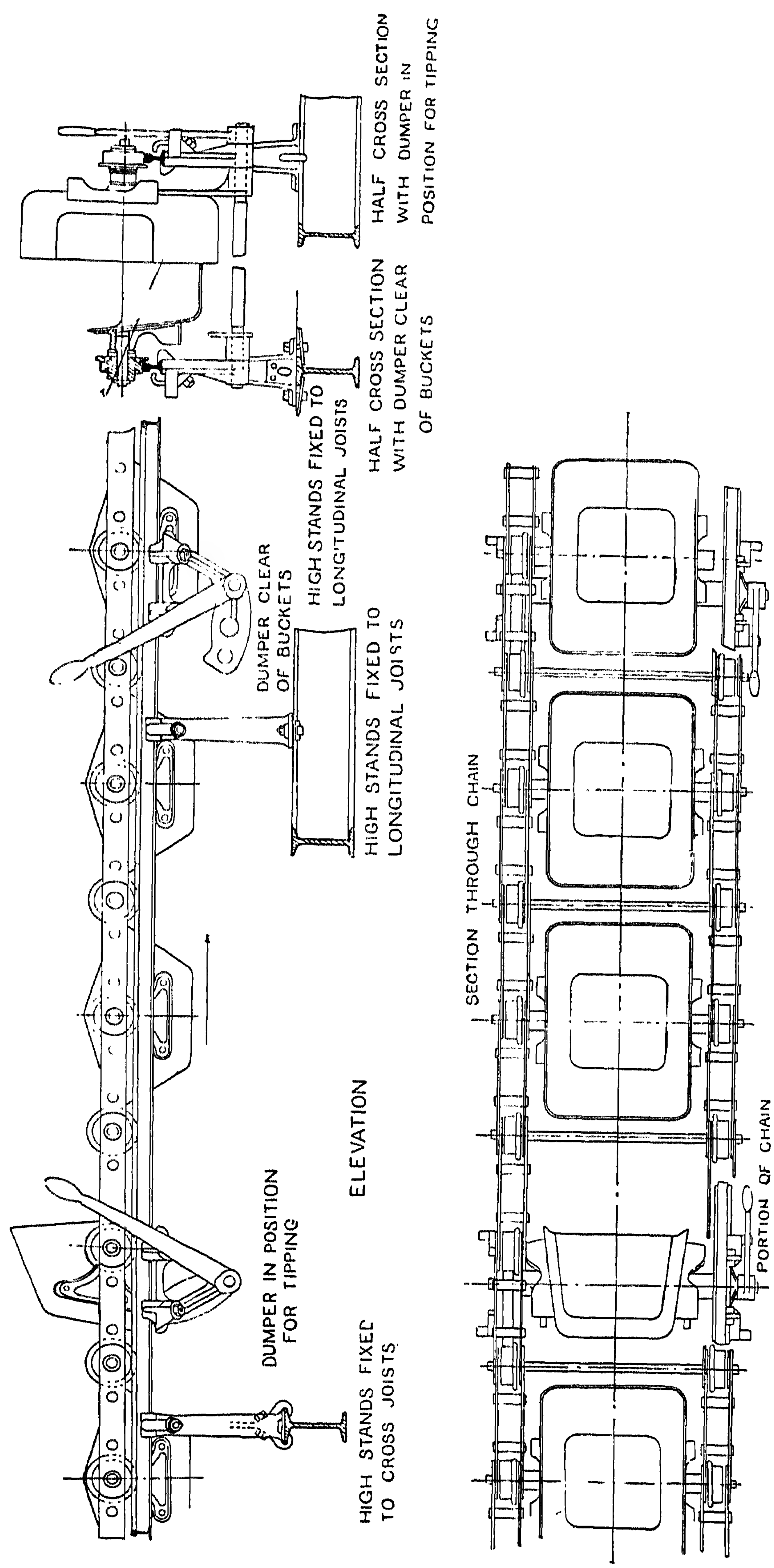

Fig. 4.—General Arrangement of Buckets, Chain, Dumpers, &c.

Illustration shows the general construction of the stamped-steel buckets, dumpers, chain wheels, &c., and also shows the methods of attaching the stands to their foundations

coal is fed into a rotary coal filler D, which automatically fills each conveyor bucket with a full complement of coal without spilling any over the sides of the bucket. The conveyor chain E passes under the rotary filler already

referred to, and, after passing round a series of guide wheels F, is carried vertically on to the conveyor gantry G, along which it travels to the boiler house. The object of carrying the conveyor over the gantry G is to allow of a store of coal being made at H for emergency requirements, such as when

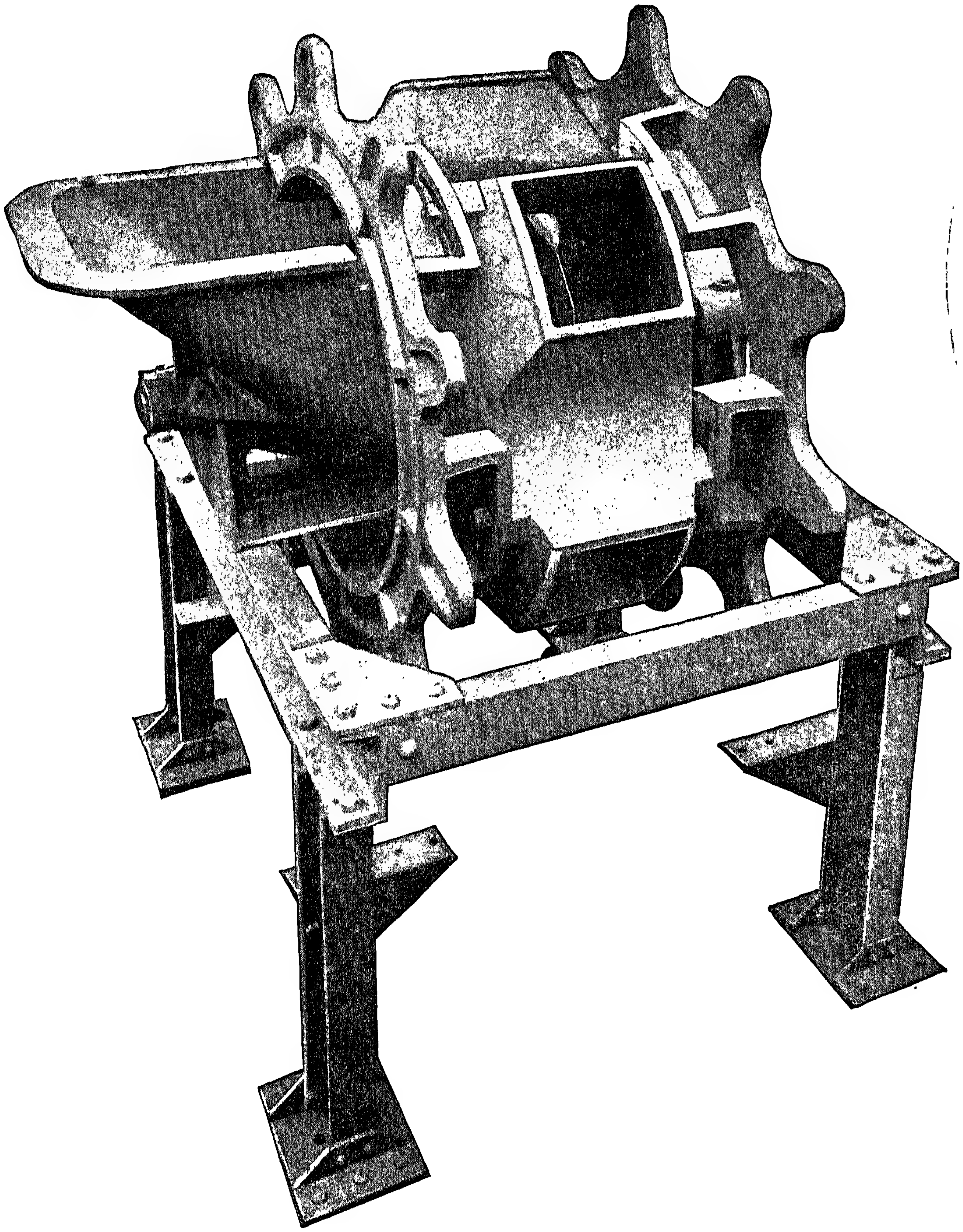

Fig. 5.—Automatic Rotary Filler

the supply of coal by rail is interrupted. Suitable arrangements are made at various points on the gantry G for tipping each bucket, causing it to discharge its coal on to the store heap. On entering the boiler house the conveyor again travels vertically until it reaches the level of the top of the coal bunkers J, along which it passes to the remote end of the boiler house.

As in the case of the gantry G, arrangements are made along the top of

the bunkers to tip the buckets at any desired point to fill each coal bunker.

As soon as the buckets are empty, and the end of the line of bunkers is reached, the conveyor descends vertically inside the end of the boiler house to the basement under the boilers, where it passes through a tunnel K under the coal store to the coal filler, thus completing one cycle.

The chain is driven by a special driving gear operated by electric motor or steam-engine at L.

Correct tension on the chain is maintained by the adjustable guide wheels F^1 in the coal filler pit.

In order to enable the store of coal to be used in emergency, arrangements are made so that coal may be drawn therefrom by way of the coal

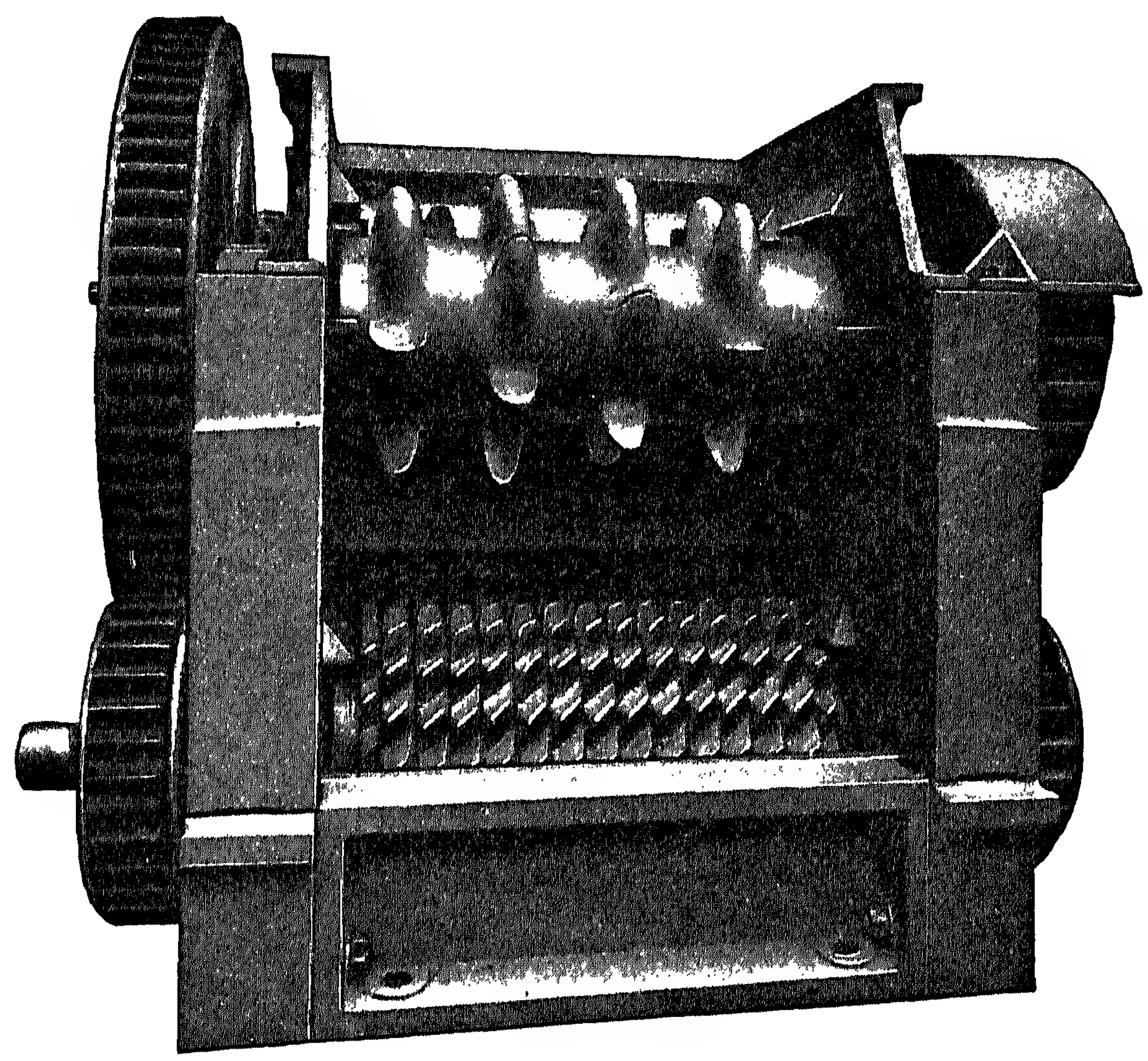

Fig. 6.—View with Part of Casing removed, showing Breaker Claws

shoots M, located under the store in the tunnel K, and, in order to efficiently fill the buckets, a portable rotary coal filler, generally similar in design to the main coal filler D in the filler pit, is provided in the tunnel. This filler is mounted on rails, so that it can readily be moved under any of the coal shoots M.

A typical design of first class gravity bucket conveyor is made by Messrs. Babcock & Wilcox, who make this apparatus in standard sizes capable of handling 20, 40, 50, 80, or 100 tons and upwards of coal per hour.

The general design and arrangement of the conveyor is shown in fig. 4, which indicates the buckets, chain, and dumpers for tipping the buckets when required. The buckets are of mild steel stamped out of one sheet, with tipping cams and trunnions riveted on.

An enlarged view of the rotary filler, indicated at D in fig. 3, is given in fig. 5. It will be noted that the device consists of a hollow casting, into the centre of which the coal is fed from the crusher and screen. The periphery

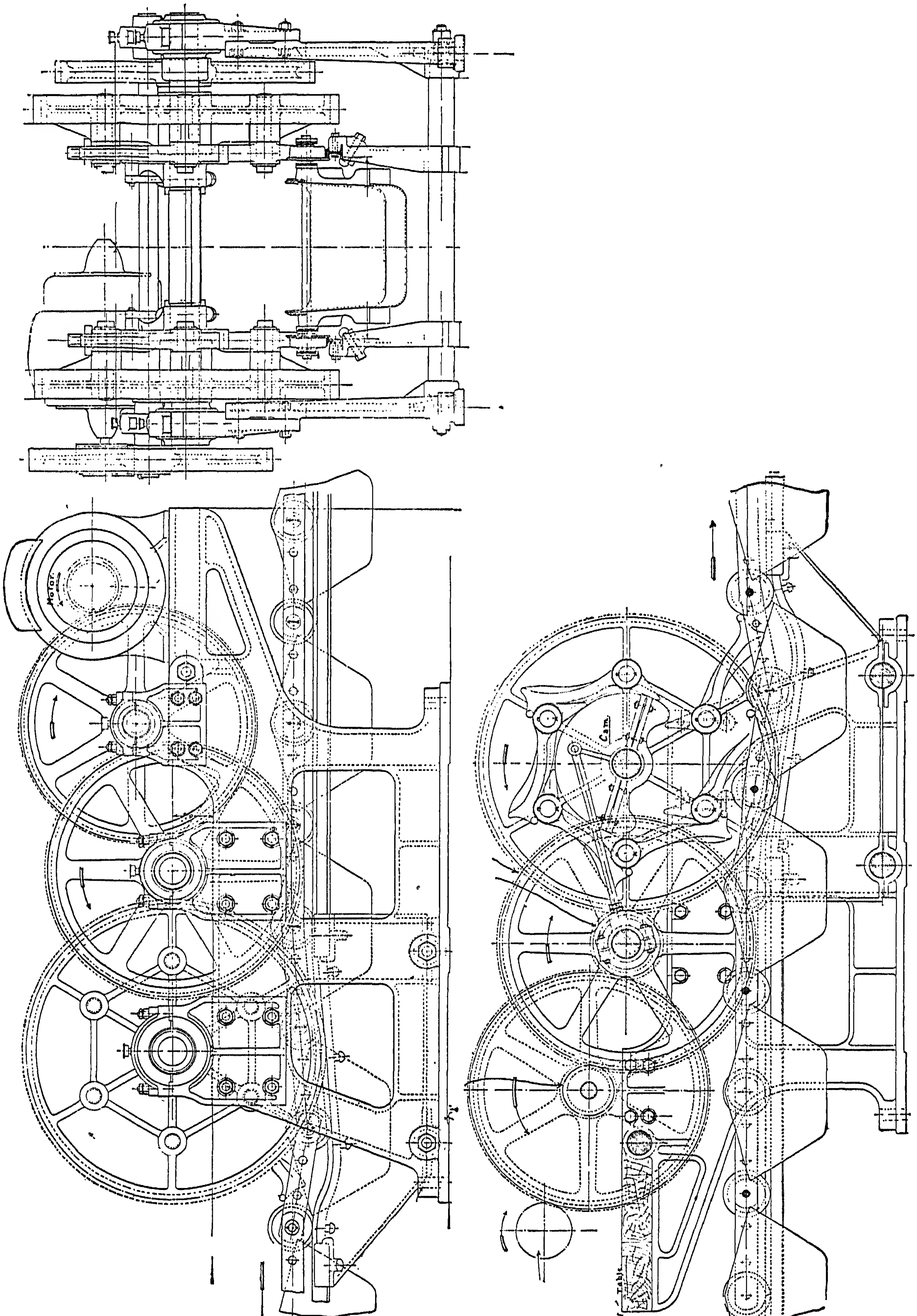

Fig. 7.—Conveyor Drive

is provided with five openings arranged to coincide with the conveyor buckets when the filler is rotated. The rotation of the filler is effected by the teeth on the sprocket wheel (seen on the side of the filler) engaging with the links of the conveyor chain. It will be seen that no separate driving mechanism

is required for the filler, and the five openings successively cover the buckets of the conveyor and fill them without spilling.

The coal crusher, indicated at C in fig. 3, is shown in larger detail in fig. 6. This apparatus is necessary for crushing any large pieces of coal received, so that the whole of the coal may be suitable for use in mechanical stokers. The crusher is generally driven by a separate electric motor, and a safety shearing coupling should be provided to guard against damage to the rolls and gearing in the event of any iron or other hard substance being accidentally delivered with the coal.

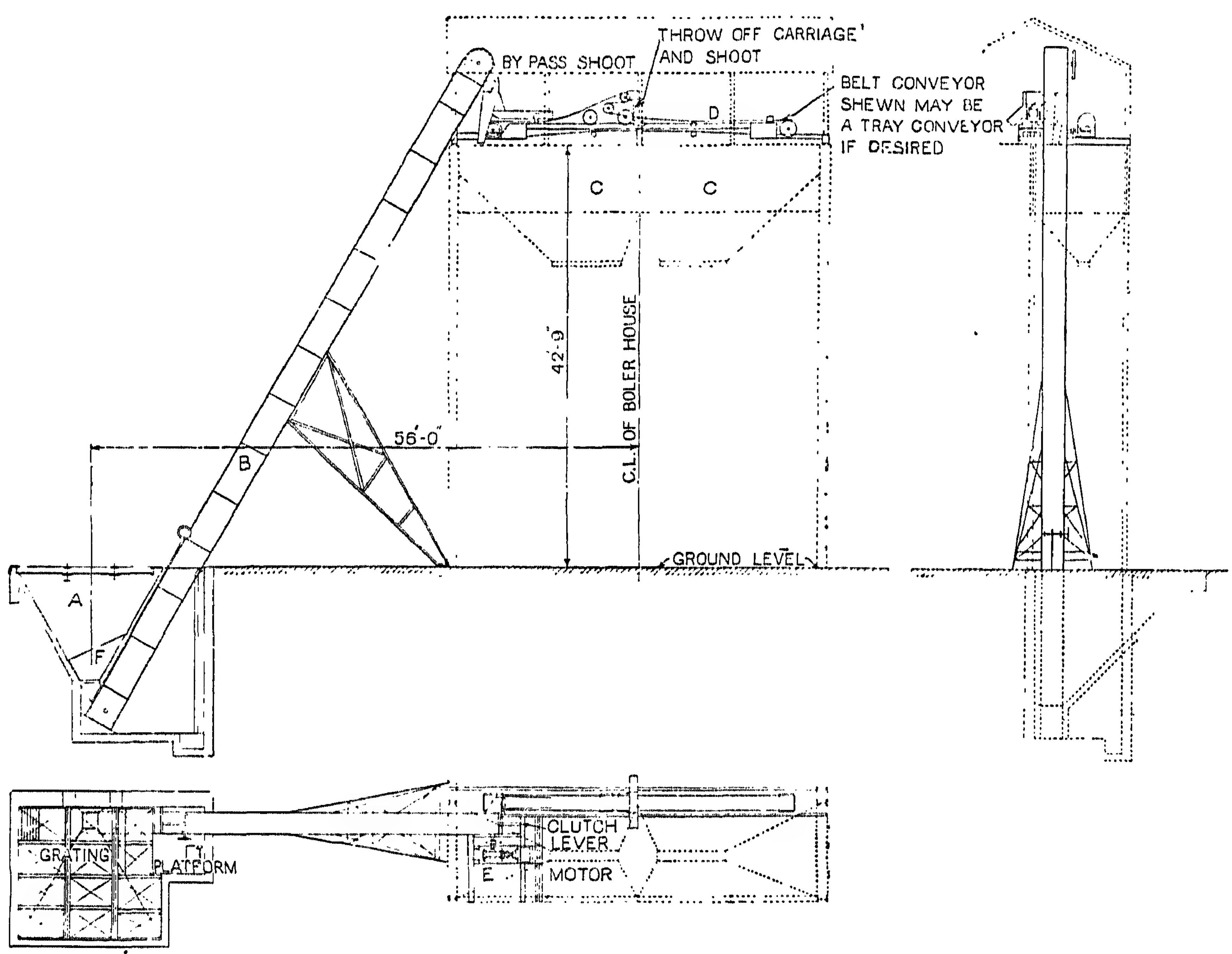

Fig. 8.—General Arrangement of Coal Handling Plant

The driving mechanism, indicated at L in fig. 3, is shown in detail in fig. 7. The special feature of this gear is the arrangement of the two sets of pawls, which successively thrust the chain in the direction of its travel and engage both chains simultaneously.

By this device the wear on the chains is compensated, and a uniform drive is obtained. It will be appreciated that some such driving device is desirable, as a plain five- or six-sided driving sprocket gives a very irregular motion to the chain, thus causing large stresses to be set up with consequent rapid deterioration of the chain and other gear.

Bucket Chain Elevator with Belt or Tray Conveyor.—A typical general arrangement of a coal handling system, in which the above combination is used, is shown in fig. 8, in which A is the coal-receiving hopper, B a bucket chain elevator for lifting the coal above the level of the coal

bunker C, and D is a belt conveyor to distribute the coal to the various bunkers. The elevator and conveyor are driven by motor and reduction gearing at E.

The elevator consists of a steel casing constructed of angles and plates, inside which travels an endless chain to which buckets are secured at intervals, as shown in fig. 9.

The chain runs over an upper or driving sprocket at the top end of the casing, while at the lower end a movable pulley is arranged for maintaining the chain in proper tension. The buckets are provided with skid bars, which slide on angle guides fixed inside the casing. The elevator may be driven by electric motor through worm and chain reduction gearing located near the top of the elevator, as at E in fig. 8. The rate at which coal is fed into the elevator may be controlled by the slide valve, as at F in fig. 8, at the bottom of the coal-receiving hopper.

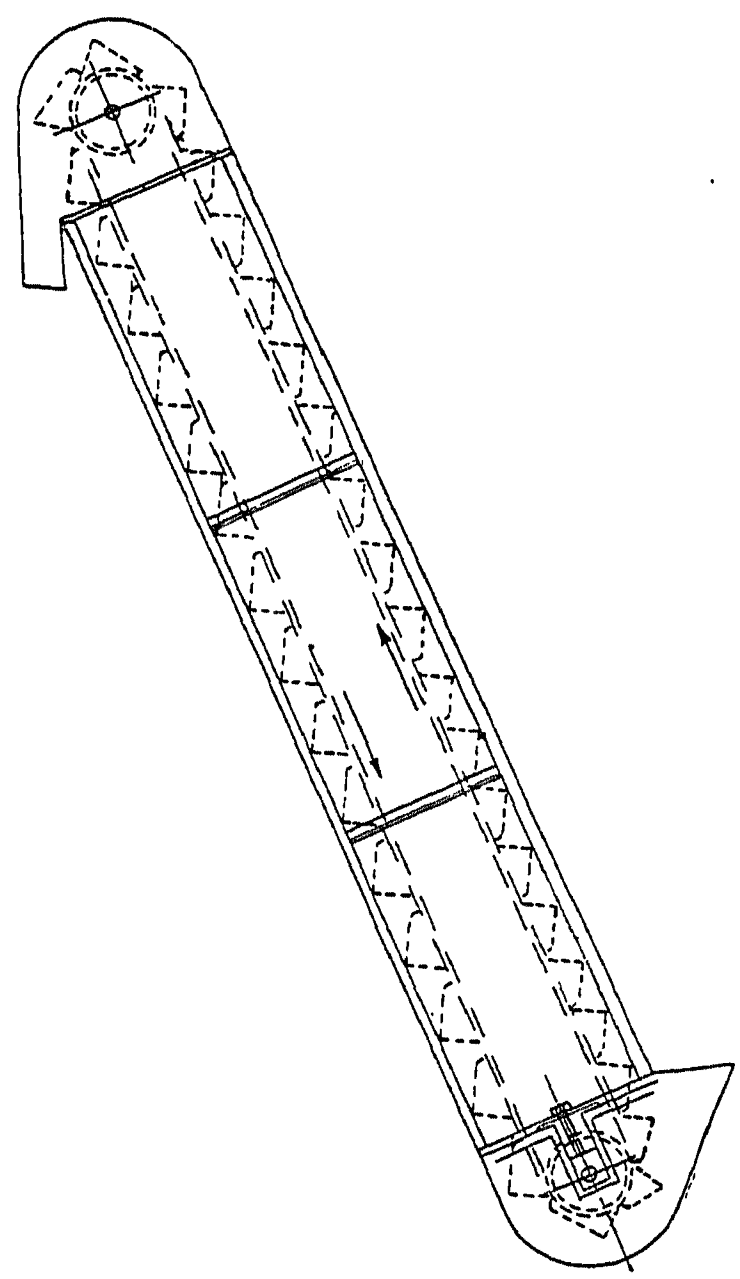
Fig. 9.—Bucket Chain Elevator

The coal-receiving hopper, shown at A in fig. 8, may be constructed of reinforced concrete, which is a cheaper construction than the mild-steel hopper shown in fig. 3.

At the upper end of the elevator the coal is tipped out of the buckets as they pass over the upper sprocket on to a belt conveyor. Alternatively the horizontal conveyor may be of the " tray " type, as described below.

Two types of tray conveyor are manufactured, one known as the tray type, in which a series of flat trays with upturned sides are attached to side chains, similar to those fitted to gravity bucket conveyors. A typical tray conveyor is shown in fig. 10, from which the main features can be seen. It will be noted that the material conveyed can only be discharged over the end of the conveyor, and it will readily be understood that this type of apparatus is unsuitable for the coal-handling system shown in fig. 8, where the coal has to be distributed at various points along the length of the horizontal conveyor into the bunkers below. The tray type of conveyor can, in certain cases, be used for elevating coal by inclining the casing at an angle of about 30° from the horizontal. In order to prevent the coal running back down the conveyor, each tray is provided with a flange across its lower edge.

In order to distribute coal at various points along the length of the conveyor, to meet the requirements of the case shown in fig. 8, another form of tray conveyor, known as the " tipping tray " type, must be used. An example of a tipping tray conveyor, as manufactured by Messrs. Babcock &

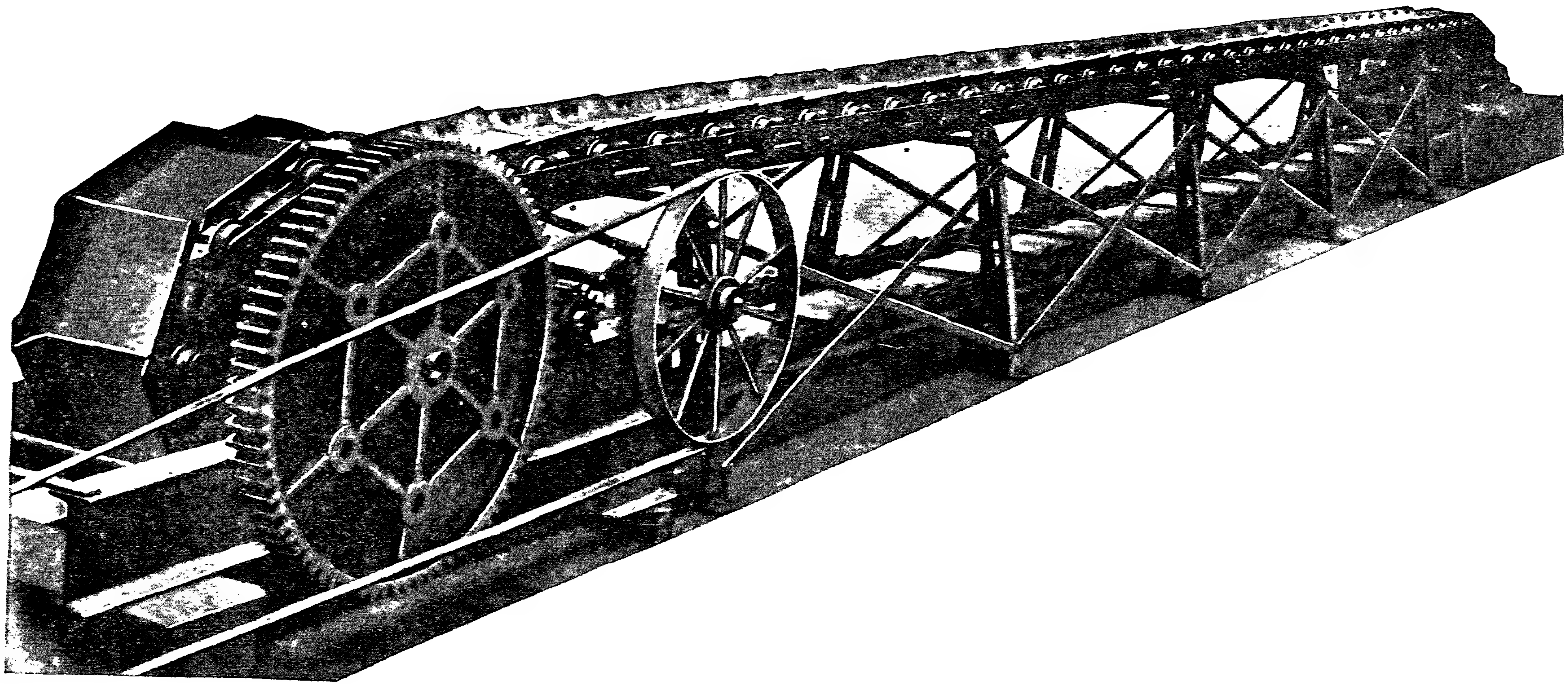

Fig. 10.—Babcock & Wilcox Tray Conveyor

Wilcox, is shown in fig. 11. It will be seen that the trays are pivoted at one end to the side chains, and, by the action of the cam strips A, the trays are tipped when required to discharge the coal at any point along the length of the conveyor. The lower line of trays hang vertically by their own weight, and allow the coal from the upper trays to fall through freely.

The side chains are similar to those used on the gravity bucket conveyor, and, as the chain is carried entirely on wheels, the power required to overcome the friction of the conveyor is small. For example, about 250 ft. of horizontal conveyor carrying 20 tons of coal per hour can be driven by a 1-h.p. motor.

Tray conveyors are manufactured in standard sizes to handle 10, 20, 40, and 80 tons of coal per hour and upwards.

Referring again to fig. 8, the horizontal

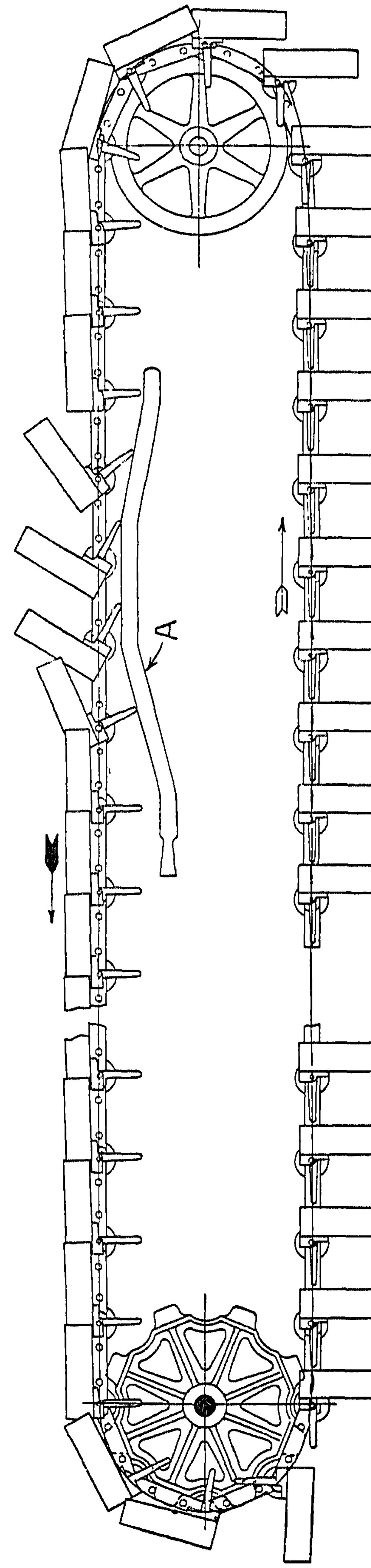

Fig. 11.—Diagrammatic Arrangement of Babcock & Wilcox Patent Tipping Tray Conveyor, illustrating its Main Features (Side elevation)

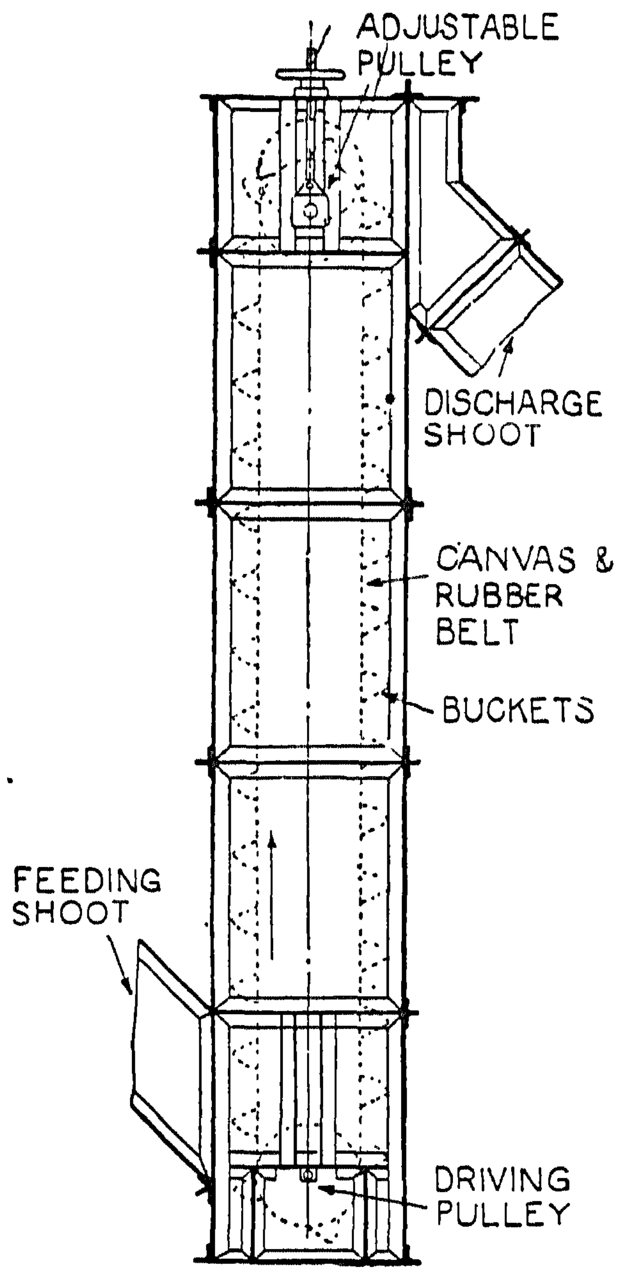

Fig. 12.—Belt and Bucket Type of Elevator

conveyor may be of the belt conveyor type as shown, instead of the tipping tray type. A general description of belt conveyors is given later.

Messrs. Fraser & Chalmers, who are associated with the Robins Conveying Belt Company of New York, U.S.A., manufacture another type of bucket elevator, which consists of steel buckets bolted to a specially con-

structed rubber belt. The belt runs over two pulleys fixed at each end of a casing, which may be of the open type or completely enclosed, as shown in fig. 12. In this particular type of elevator the adjusting gear for maintaining the correct tension on the belt is located at the upper end of the casing. This is an important feature, as it permits the lower pulley to remain in a fixed position, thus maintaining a constant clearance between the buckets and the curved bottom plate of the boot.

These elevators have been built with the upper and lower pulleys spaced 95 ft. apart, and with a handling capacity of 200 tons per hour. It will be appreciated that the belt and bucket type of elevator has an advantage, from an operating point of view, over the metal chain and bucket type, in that a chain which breaks during working usually does so without warning, whereas an elevator belt always gives ample warning before failing, so that a replace belt may be obtained or repairs carried out at the first opportunity.

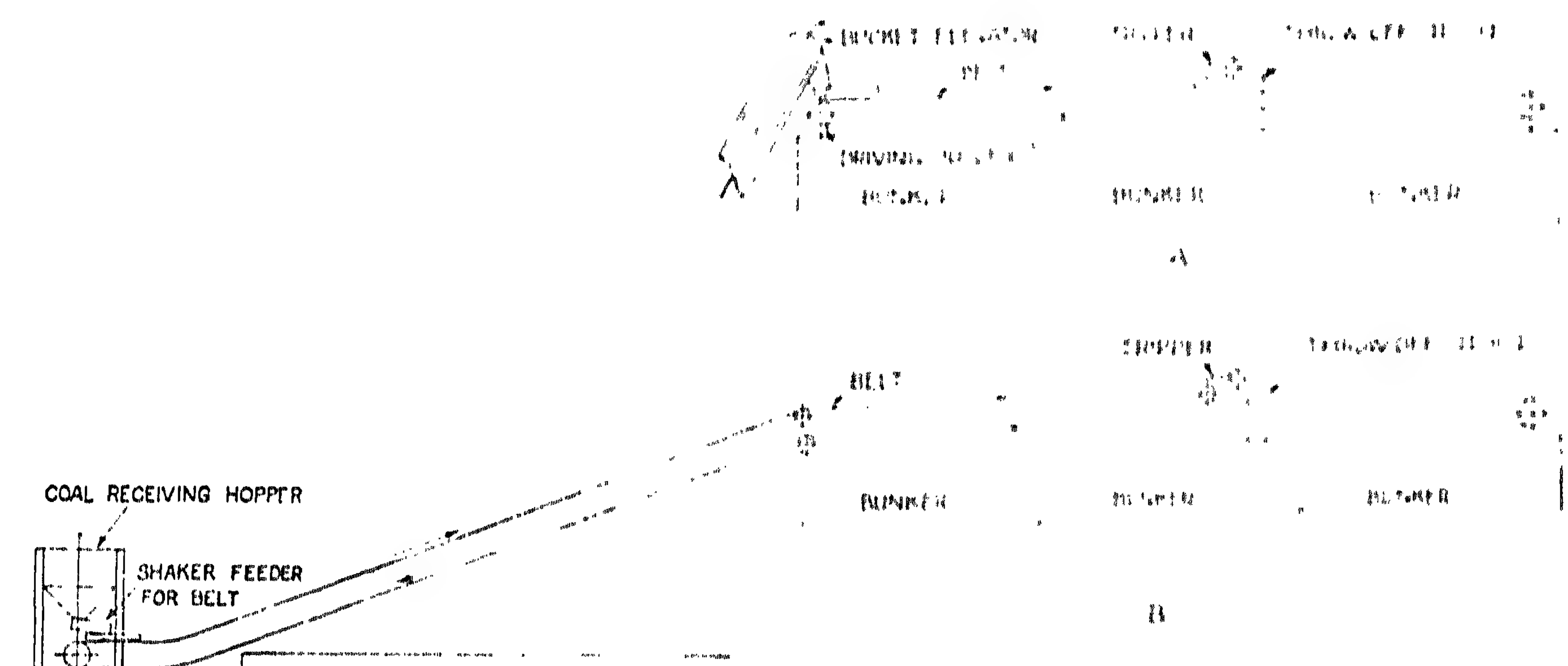

Fig. 13.—Typical arrangements of Belt Conveying Systems

Belt Conveyors.—It is claimed that the belt conveyor has many advantages over the chain and bucket types of conveyors in that the belt conveyor is simpler and can be made in larger capacities. It is lighter and therefore requires a cheaper and more simple structure to carry it. Further, it is less likely to fail without giving warning, as, unlike a metal chain, the belt shows unmistakable signs of impending failure some considerable time beforehand, and provision for replacement or repairs can be made.

Typical arrangements of belt conveying systems are shown in fig. 13, in which diagram A shows a simple horizontal belt conveyor, as in fig. 8, arranged for receiving coal from a bucket elevator and distributing it to the coal bunkers underneath. Diagram B shows a complete coal handling system for a power plant, consisting of one belt elevator conveyor arranged to carry coal from a coal receiving hopper situated below rail level to the overhead bunkers. The elevating portion of the belt is inclined at an angle of about 18° to the horizontal, so that the coal will not run back down the belt.

In each case a travelling tripper is provided on the top of the bunkers for throwing off the coal where desired.

A first-class belt conveying plant is made by Messrs. Fraser & Chalmers Engineering Works, who are associated with the Robins Conveying Belt Company. The conveyors are made in sizes up to and over 800 tons per hour in a single belt, and the following illustrations show details of these belt conveyors.

The belt is composed of cotton duck and rubber, and is known as the Robins Patent "Stepped Ply" Belt. The belt is constructed with more

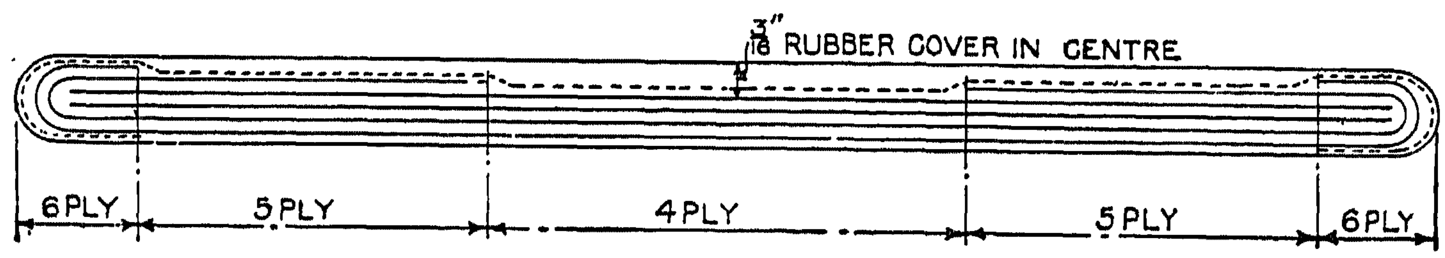

Fig. 14.—Cross Section showing Construction of Robins Patent "Stepped Ply" Belt

plies of duck at the edges than at the centre, as shown in fig. 14. This arrangement has many advantages over the ordinary straight ply belt, in that it increases the life of the belt and increases the flexibility, as it allows of a thicker rubber covering being provided in the centre, where the greatest wear takes place. The sides of the belt are also strengthened, by reason of the additional plies of duck, while the edges of the belt are covered continuously right round with a protective coating of rubber which protects the duck from wear.

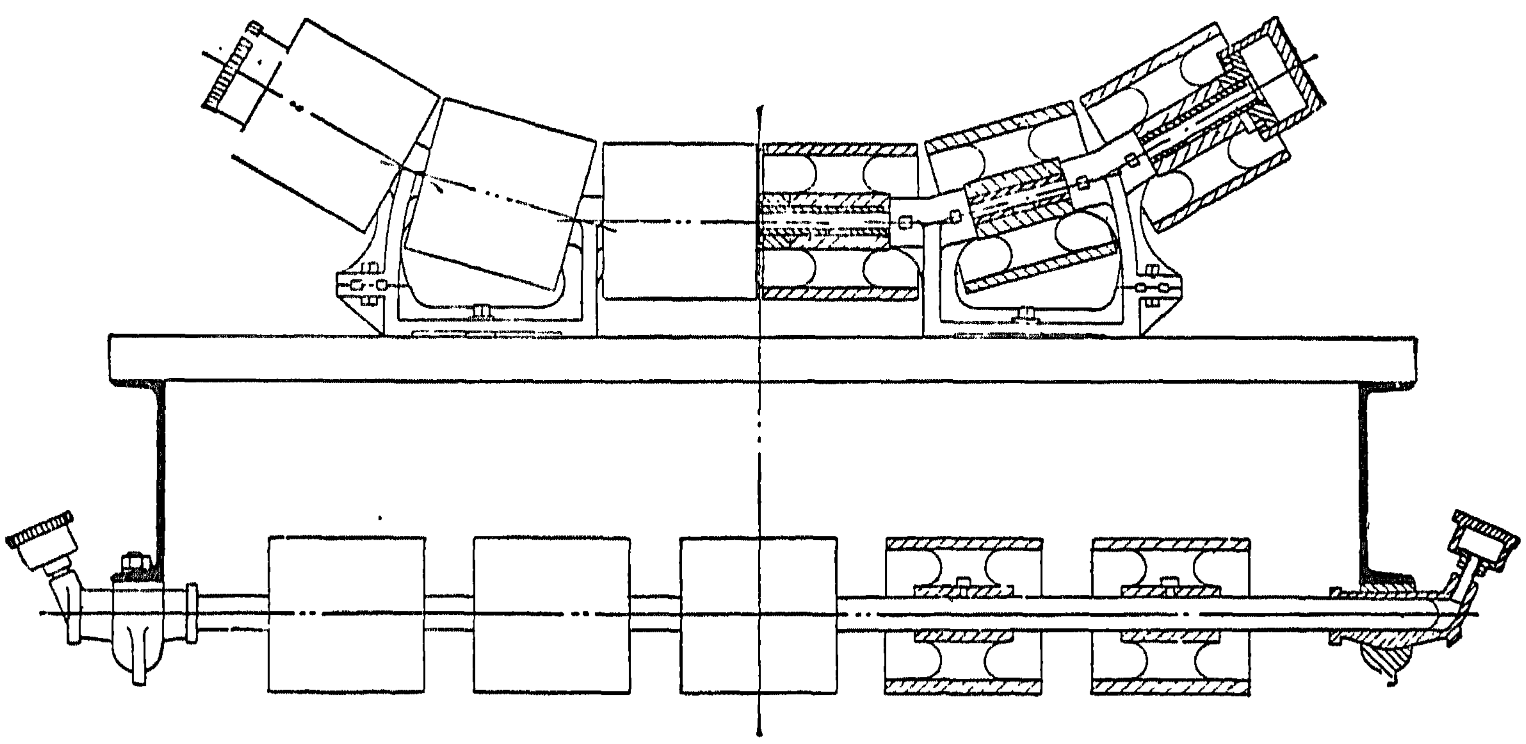

Fig. 15.—Robins Standard Troughing and Return Idlers

The belt is carried over pulleys and idlers, mounted on a framework of timber and steel. The upper idlers are curved to cause the belt to take a trough form to carry the coal, while the lower or return idlers are straight.

Fig. 15 shows a 5-shaft idler arrangement, which is known as the Robins Patent Troughing and Return Idler. This type of idler is manufactured for belts up to 60 in. wide, and causes less wear on the belt than the more usual 3-shaft type. In certain cases, where the friction and weight of the revolving parts are of importance, ball or roller bearings are fitted to the idlers. Grease lubrication, fed through the idler shafts, is used on both types of idler.

The end bend and tripper pulleys are of strong construction, and turned and balanced to eliminate vibration. When necessary the driving pulleys are coated with rubber to give better driving power. The rubber coating is securely attached by vulcanizing the rubber under pressure to a copper-plated pulley.

The correct tension in the conveyor belt is maintained by adjusting the position of one of the end pulleys. This is generally effected by mounting

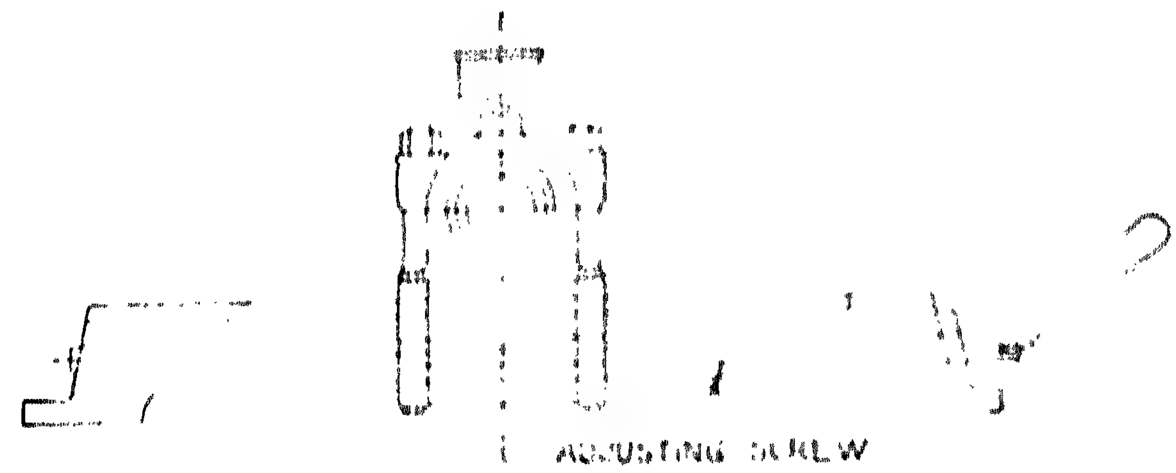

Fig. 16.—Robins Protected Screw Take-ups

the adjusting pulley shaft bearings on slides, controlled by screws so that the tension can be adjusted. The slide, which is shown in fig. 16, is graduated in divisions of $\frac{1}{8}$ in., so that equal adjustment in both screws is easily ensured. In certain cases, such as where a belt conveyor is hinged, which causes a variation in the length of the belt, weight controlled tension gear is provided, which automatically keeps the correct tension on the belt.

The coal conveyed by the belt can be discharged at the head end of

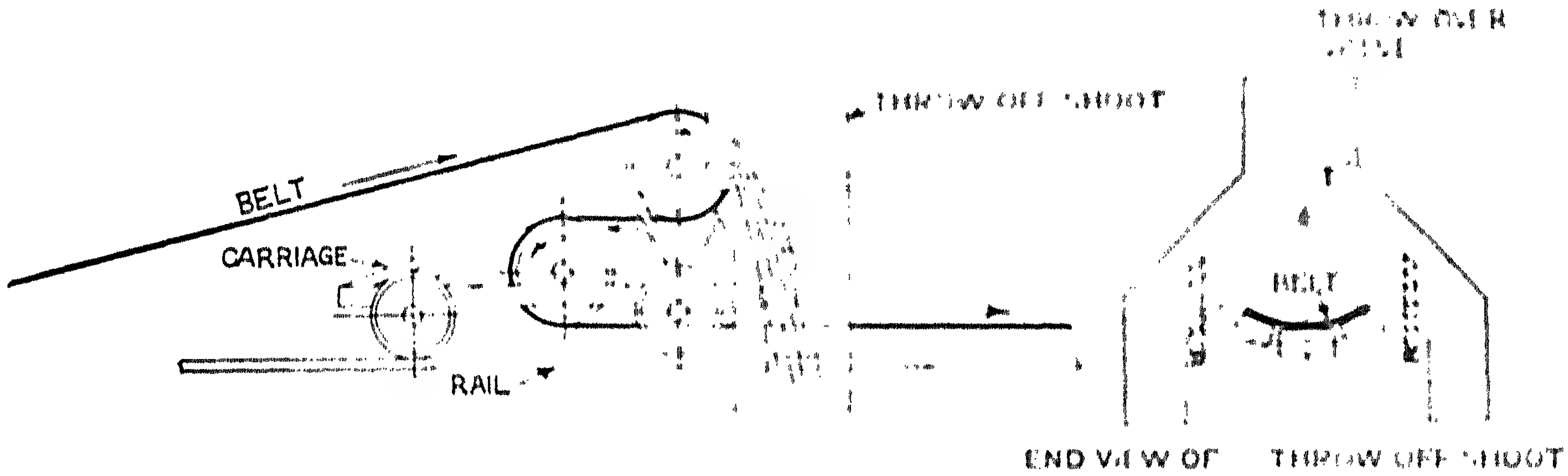

Fig. 17.—Travelling Tripper

the conveyor or at any intermediate point by the installation of a fixed or travelling tripper.

The general features of a travelling tripper are shown in fig. 17, from which it will be seen that the apparatus consists of a framework running on wheels mounted on a track, and carrying two pulleys over which the belt passes. A double shoot is provided immediately in front of the upper pulley, and the whole arrangement is such that the coal on the belt is thrown into the shoot as the belt passes round the upper pulley. The path of the belt is shown in fig. 17. The double shoot may be fitted with a throw-over valve to direct the stream of coal to either side as desired. Travel-

ling trippers are either propelled by hand or are automatically traversed backwards and forwards between any desired limiting points on the conveyor length. The automatic tripper is operated by gearing actuated by the conveying belt itself, so that no separate driving mechanism is required. With the automatic tripper the coal is evenly discharged, as the tripper moves backwards and forwards along the length of its track and requires no attendance.

In the arrangement shown in fig. 8, where the belt elevator is fed from a bucket elevator, it will be seen that the coal is automatically discharged in small quantities on to the belt, and that no special precautions need be taken

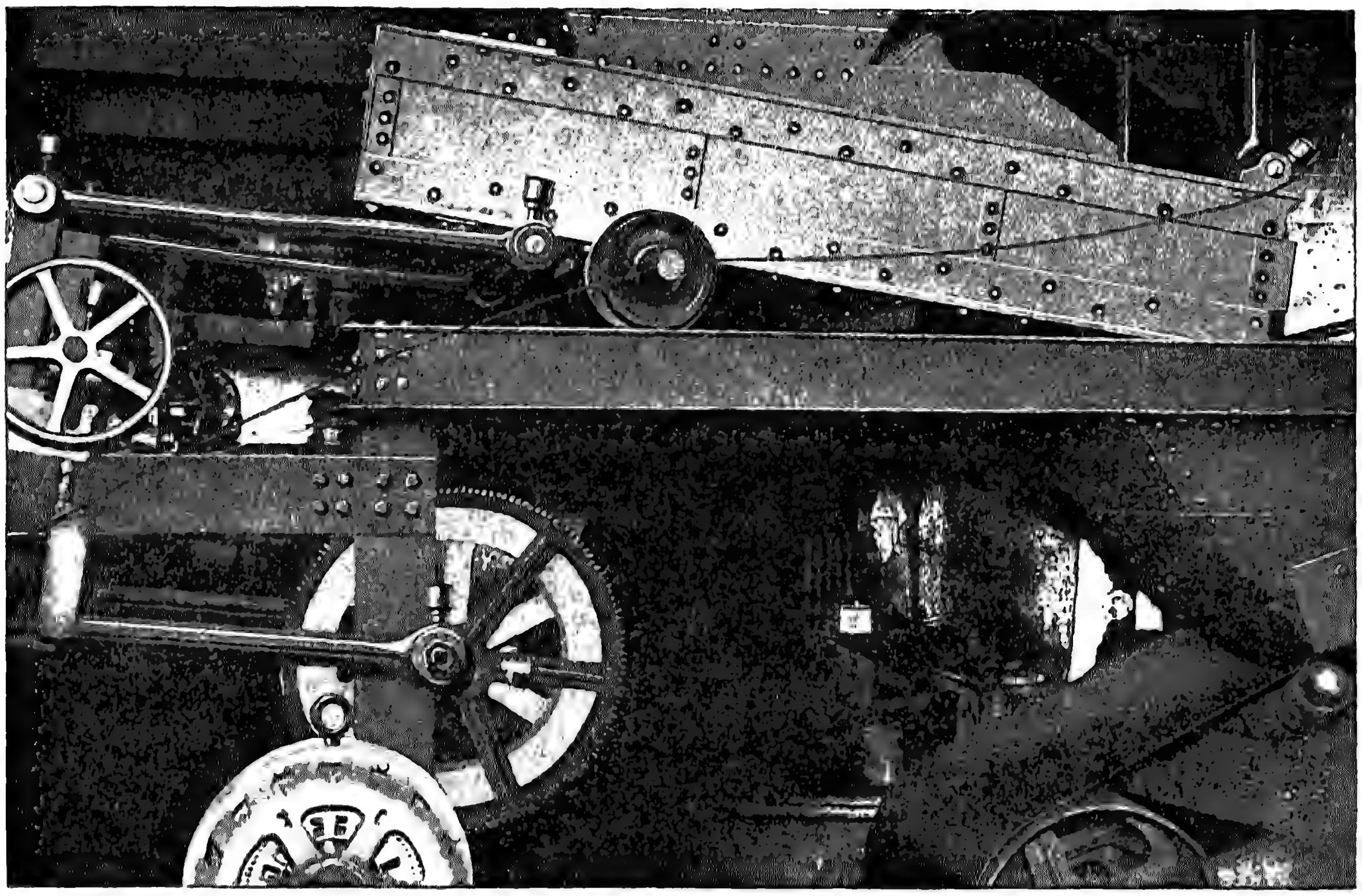

Fig. 18.—Shaking Belt Feeder

to ensure uniform loading of the belt. When feeding a belt from a storage bunker or coal-receiving hopper, however, it is necessary to provide means for controlling the flow of coal on to the belt.

It will be appreciated that the wear of the belt, due to the abrasion of the coal, is caused by the dropping of the coal upon the belt at the feeding-on point, as elsewhere on the belt the coal simply lies on the belt and is carried forward.

It is, therefore, important that the feeding arrangements are such that a minimum amount of abrasion of the belt takes place, and to obtain these conditions the coal must be fed uniformly and should fall on to the belt in the direction of its travel with as little shock as possible. Several types of belt conveyor feeders are used, among which may be mentioned the shaking feeder and the roll feeder.

In all cases the feeding device is designed to give a uniform flow of coal

on to the belt, and the feeder is preferably geared to the belt driving gear so that feeder and belt start and stop together.

A typical shaking feeder is shown in fig. 18, in which the rate of feed is controlled by altering the length of the stroke of the shaker and/or the angle of inclination of the pan.

The roll feeder is shown in fig. 19, and consists of a roller geared to the belt mechanism and placed immediately below the coal hopper outlet.

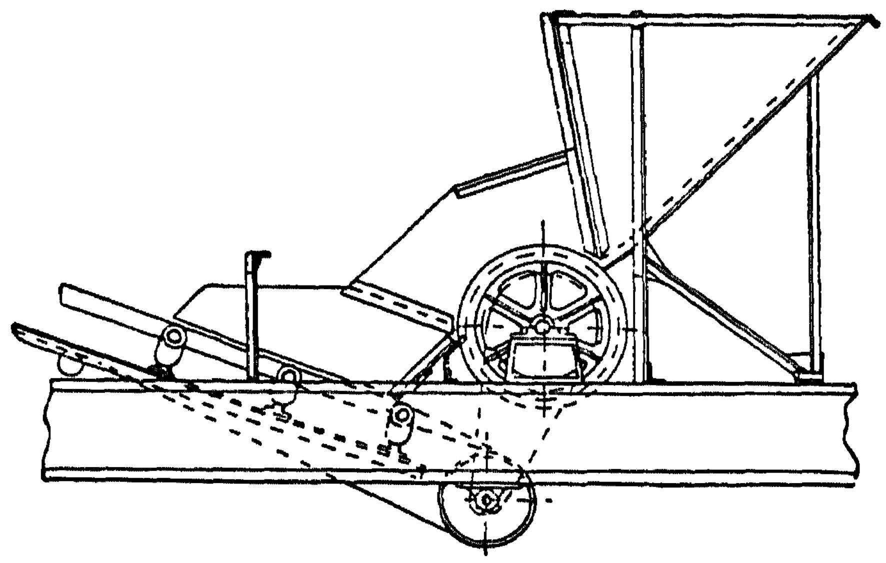

Fig. 19.—Roll Feeder

Adjustment of the coal valve varies the speed at which the coal is fed on to the belt. The coal ceases to flow immediately the roll stops.

CHAPTER III

Ash Handling

The percentage of ash in coal used in power plants has gradually increased during the last decade, until to-day it is quite common to find the percentage of ash amounting to 25 per cent and over. This increase in the ash content of the fuel, and the general increase in the size of power plants, makes the handling of ashes an important one in the economical design of the plant, and it will be found that the cost of ash handling represents a not inconsiderable part of the operating costs.*

In order to reduce these costs, therefore, it is essential to install reliable and efficient ash handling plant, and the design of such plant for handling large quantities of ashes with the minimum cost of labour and repairs has received close attention from engineers, with the result that important improvements have been made, to which reference will be made later.

It will be appreciated that ash conveying plant works under adverse conditions owing to the abrasive action of the dust, together with the heavy corrosion which is caused by the moisture and fumes which are given off when the ashes are quenched.

* See article by Sir Richard Redmayne in the "Fuel Supplement" to the *Colliery Guardian*, Vol. I, No. 3, March, 1922.

In small plants the ashes from the boiler ash hoppers can be periodically emptied into ash wagons arranged to run on a narrow-gauge railway under the ash hopper doors. After quenching the ashes with water, the wagons can be pushed along to a bucket ash elevator for lifting the ash into an ash storage bunker. Such an arrangement is shown more or less diagrammatically in fig. 20, in which the ashes from the boiler ash hoppers A are

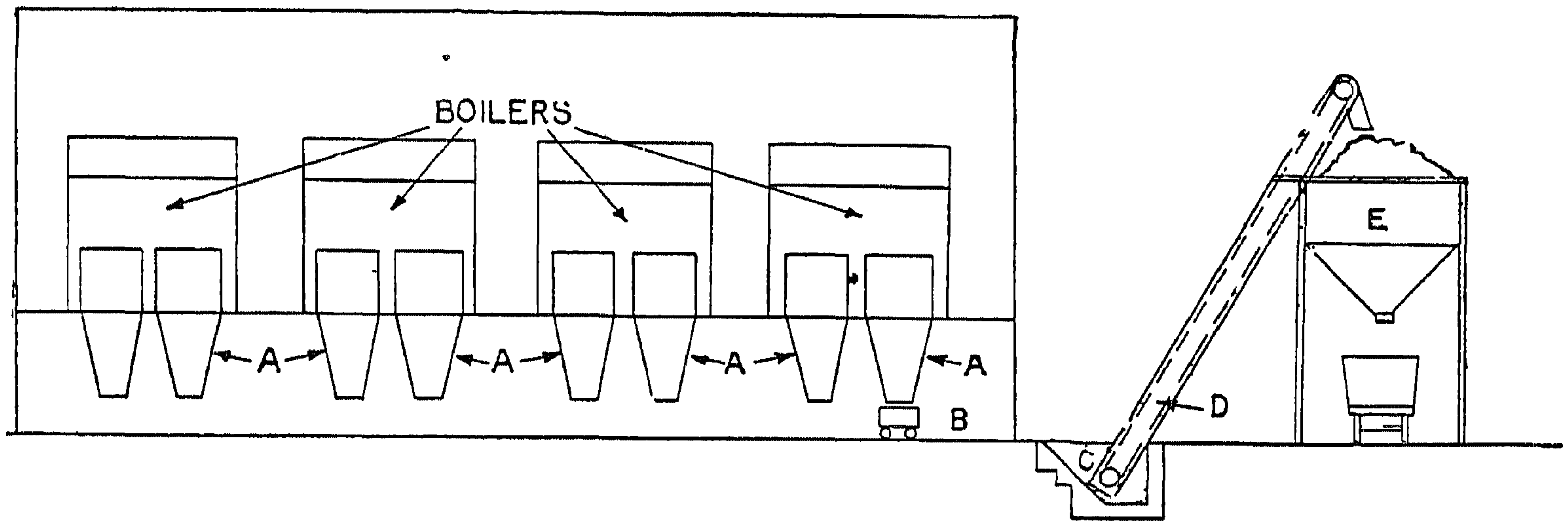

Fig. 20.—Typical Arrangement of Ash Handling Plant with Ash Wagons and Bucket Elevator

discharged into travelling narrow-gauge tipping ash wagons B, whence they are tipped at C into the boot of a bucket elevator D, which elevates the ashes to the storage bunker E, from which the ashes may be removed by road or rail.

An alternative arrangement is shown in fig. 20 A, in which provision is made to run the ash wagons B into an electric hoist C, which lifts the

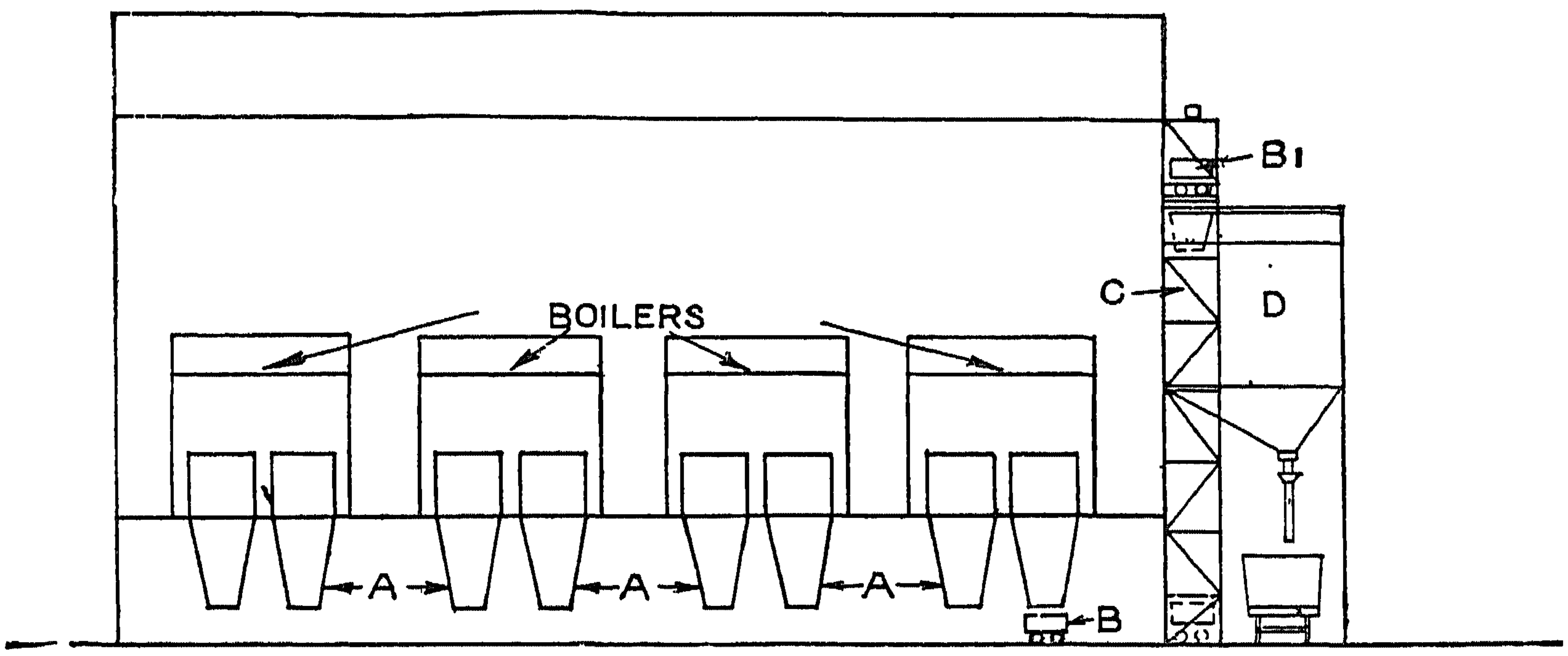

Fig. 20 A.—Typical Arrangement of Ash Handling Plant with Ash Wagons and Ash Wagon Hoist

complete ash wagon up to B_1, and tips it into the ash storage bunker D. In larger plants the ashes may be handled direct from the ash hoppers under the boilers by gravity bucket elevators or by tray conveyors, and thereby delivered into the ash storage bunker.

In certain cases it is convenient to arrange the returning length of a gravity bucket coal elevator chain to receive the ashes direct from the boilers and discharge them into the ash storage bunker, as in fig. 21.

In large stations, however, the weight of ash to be dealt with per day

is so great that it will be found more convenient and economical to have a separate ash handling system.

It will be seen that in all the above examples of ash handling plant the ashes are received in a more or less red-hot condition from the ash hoppers under the boilers, and that it is necessary to quench the ashes on the conveyors or in the ash wagons. It is inevitable, therefore, that considerable quantities of steam and dust are liberated, which are not only very destructive to any machinery and steelwork with which they come into contact, but also create an atmosphere which makes the operation of ash plant extremely trying to the men and detrimental to their health, with the result that men cannot work continuously in the ash tunnels, and it is necessary to have a large staff of men with consequent high costs.

Two notable advances in ash handling plant design have been made

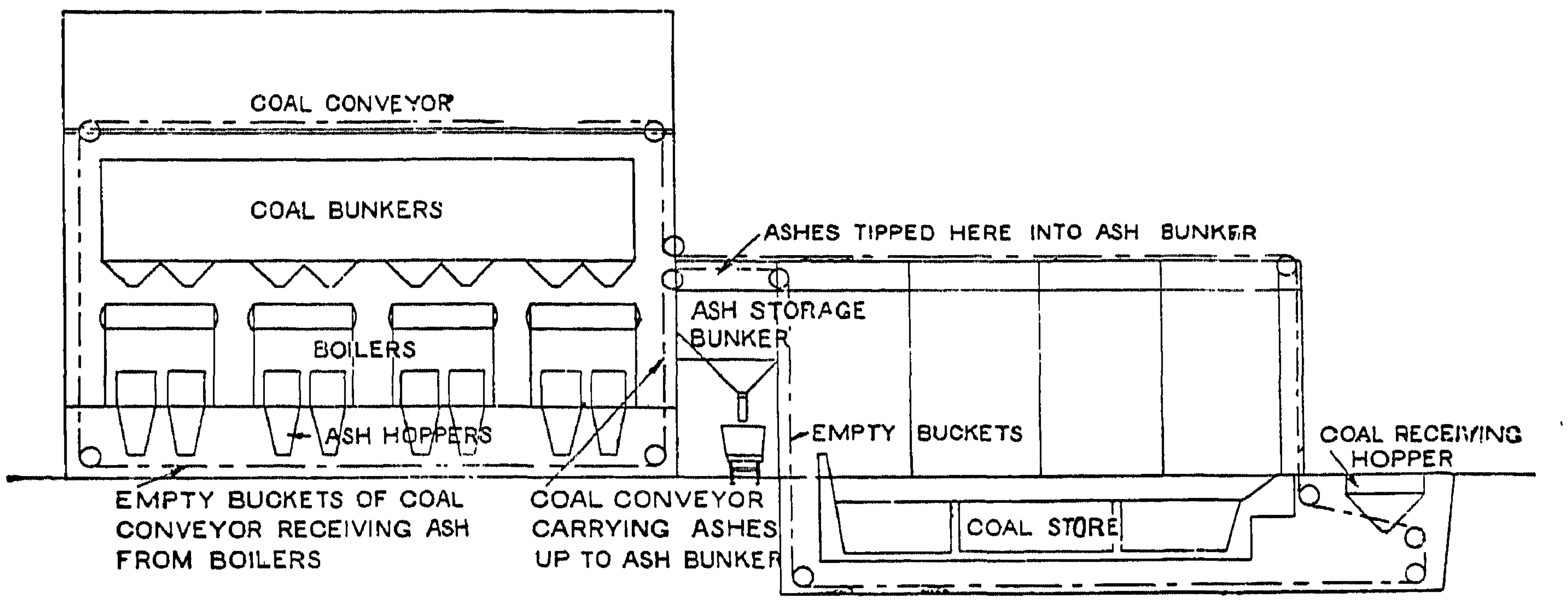

Fig. 21.—Typical Arrangement of Coal Elevator designed to handle both Coal and Ashes

comparatively recently, in which the creation of dust, fumes, and steam are entirely eliminated, and at the same time the labour required is considerably reduced.

The two designs referred to are known as (1) the Vacuum Ash System, as manufactured by Messrs. Babcock & Wilcox, Ltd.; and (2) the Water-sealed Ash Conveyor, manufactured by the Underfeed Stoker Company, Ltd.

Vacuum Ash System.—In this system the ashes are conveyed through a series of pipes by a high-velocity current of air, and thereby carried into an ash receiver from which they are discharged periodically for removal by rail or road.

A diagrammatic arrangement of a vacuum ash system is shown in fig. 22, in which the ash hoppers below the boilers are indicated at A. The ash pipes B are provided with openings under each boiler ash hopper to receive the ashes. These openings are provided with lids to close them when not in use. The end of the ash pipe B is left open to the atmosphere at C, and the main supply of air for conveying the ashes enters the system at this point. The ash pipe B is led to an ash receiving bunker E, where the ashes are delivered and quenched by water sprays J. The ash receiver is provided with

an ash discharge valve at the bottom, through which the ashes are periodically emptied into railway wagons.

The air for conveying the ashes to the ash receiving bunker is drawn through the system by a motor-driven exhauster G, by way of pipe F, and through the dust extractor G_1. The exhaust from the exhauster G is discharged through a silencer and atmospheric pipe H.

With certain coals the ash is delivered from the ash hoppers under the boilers in relatively large slabs, and it is necessary that these should be broken before being admitted into the ash pipe. For this purpose a travelling motor-driven 4-roll ash breaker is provided on rails below the ash hopper doors. This crusher is indicated at K, and the ashes are always passed through this crusher before entering the pipe. It is found in practice that no dust is created when ashing operations are in progress, as the air, which is drawn

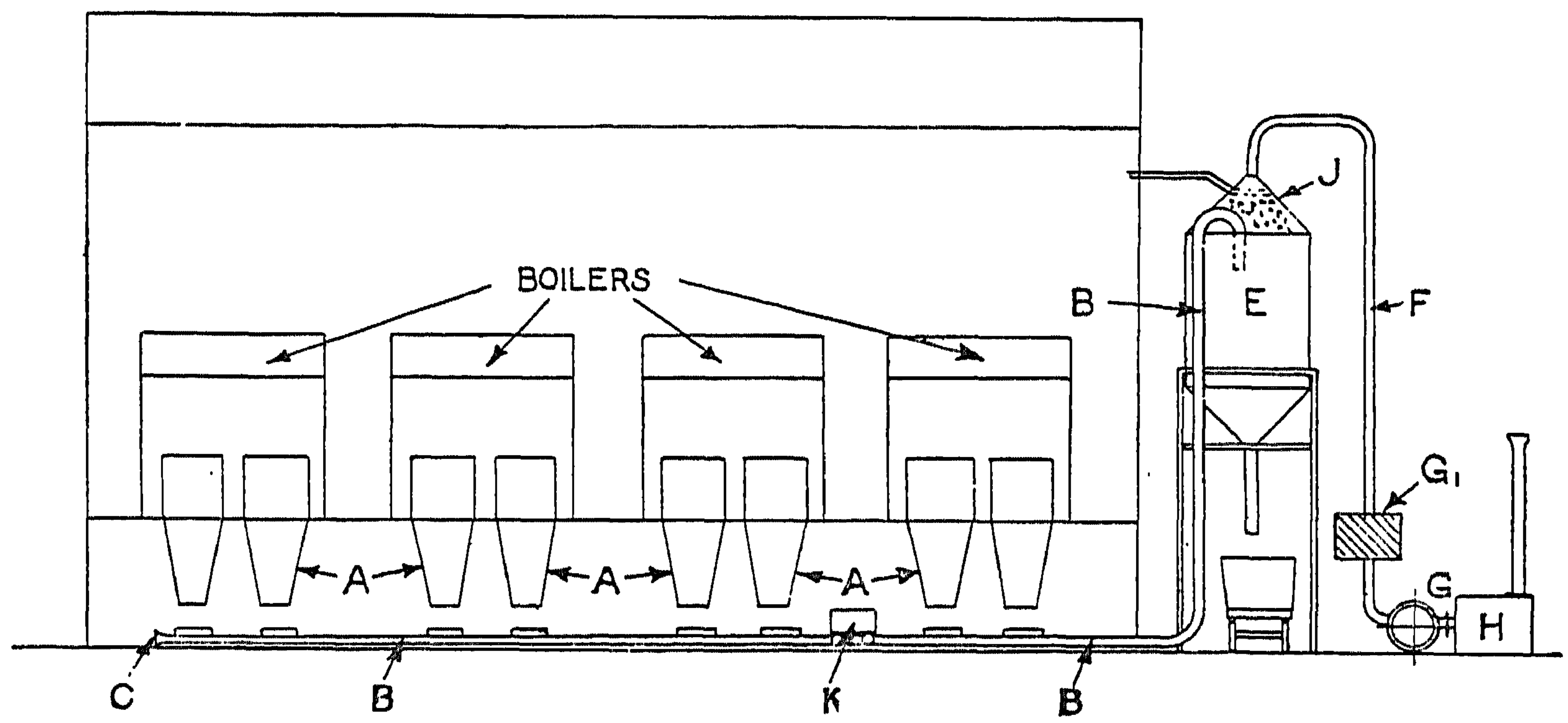

Fig. 22.—Diagrammatic Arrangement of Vacuum Ash System

through the ash crusher and through the ash pipe openings, effectively carries away all dust and fumes with the ashes.

The main supply of air, being drawn into the ash pipes through the open end C, causes a large volume of fresh air to be continually carried into the ash tunnel or basement, whereby an excellent ventilation is secured.

The travelling ash crusher already referred to is shown in fig. 23. The ash door of the ash hopper under the boiler will be seen immediately above the crusher, while on the floor behind the crusher the ash pipe will be noticed.

The vacuum ash system can be used to convey the ashes some 650 ft. from the boilers to the ash-receiving bunker. An illustration of part of a vacuum ash installation, in which the ashes are conveyed through about 350 ft. of ash pipe, is shown in fig. 24. In this instance the top of the ash-receiving bunker, seen on the right hand of the illustration, is about 45 ft. above the point at which the ashes are discharged from the boiler ash hoppers. The piping shown is 10 in. internal diameter, and the average vacuum in the receiver when ashing is in progress is from 4 to 5 in. of mercury. The exhauster is located in the small house immediately below the ash receiver, and the round object in front of the exhauster house is the silencer. In

this case an atmospheric pipe from the silencer is not fitted, and the air is simply discharged from the end of the silencer, which consists of a steel pipe fitted with baffle bricks. The ashes pass along the pipes at considerable velocity, and, as might be expected, the wear on the pipes owing to sand-blast action is considerable. It is therefore necessary to carefully consider the design of the piping system so that the maximum life of the pipes may be obtained, and, at the same time, the replacement of worn pipes can be easily and quickly carried out.

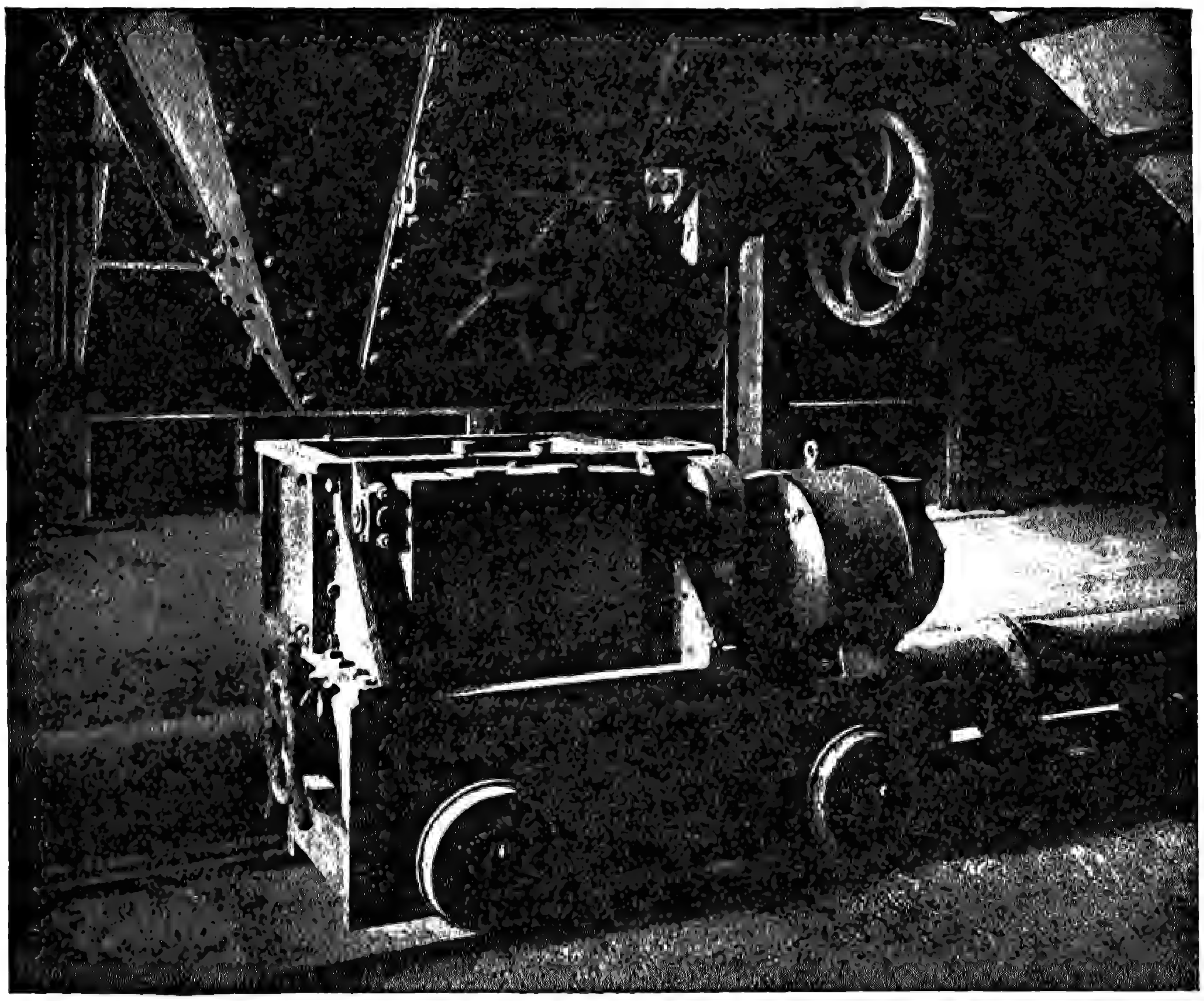

Fig. 23.—Babcox & Wilcox Patent Suction Ash Conveying Plant

View in ash tunnel, showing breaker and ash pipe

The wear on the pipes is, of course, most rapid at the bends, and therefore the general run of piping should be kept as straight as possible and all right-angle bends avoided.

In order to make the cost of replacements as low as possible, standard interchangeable pipes should be used; for example, all the straight pipes should be of one size and the different bends should be limited to two or three patterns, as by adopting this arrangement the provision of spare pipes is most easily arranged.

The pipes should be plain double spigot pipes of cast iron of considerable thickness, say $1\frac{1}{2}$ in. for the 10 in. internal diameter pipes, and the joints should consist of cast-iron double socket collars (Kimberley collars) packed with asbestos rope. The joints do not require heavily caulking up, as, for

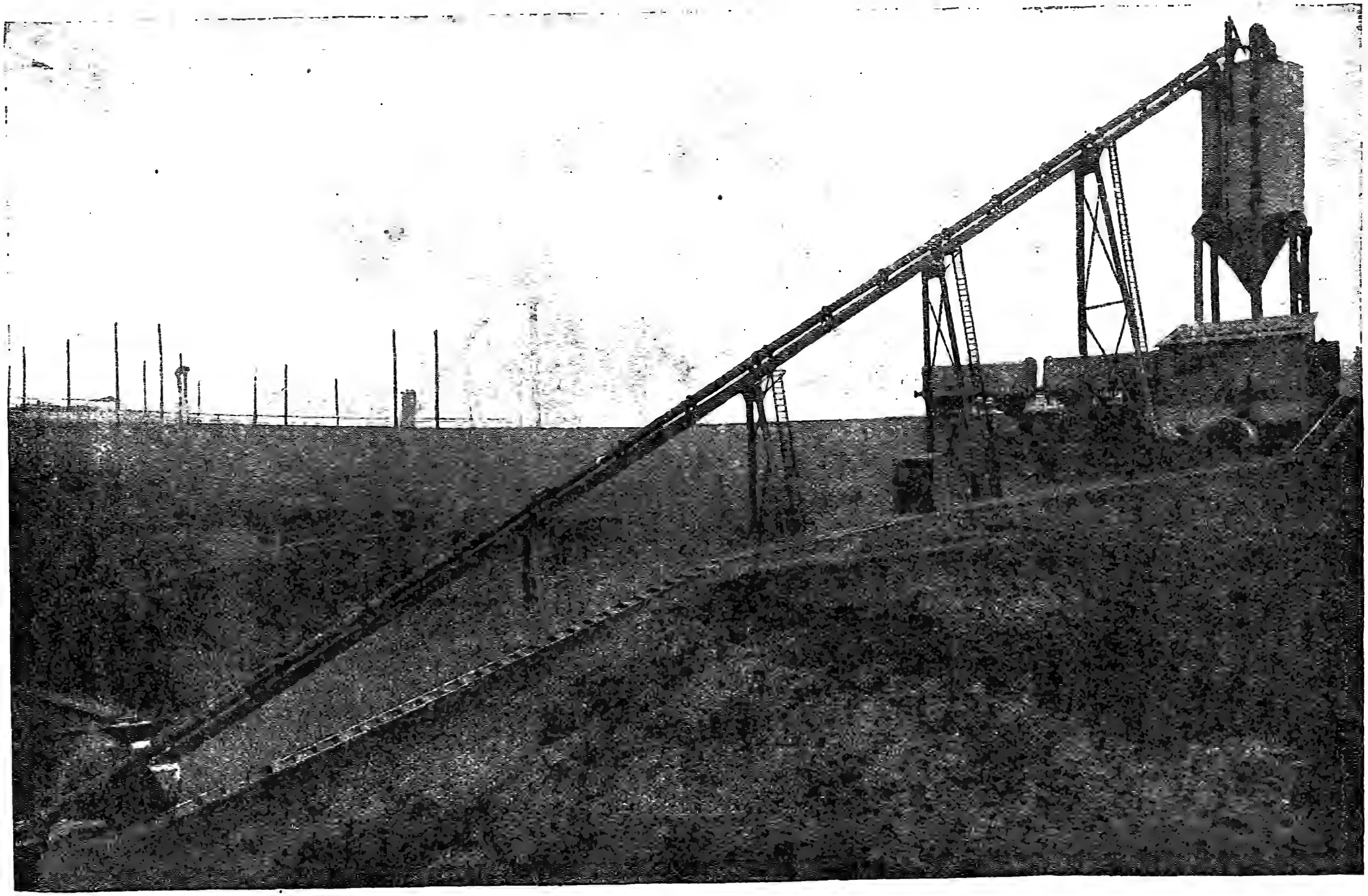

Fig. 24.—View of Ash Receiver and Connecting Piping of Plant handling 10 tons of ashes per hour

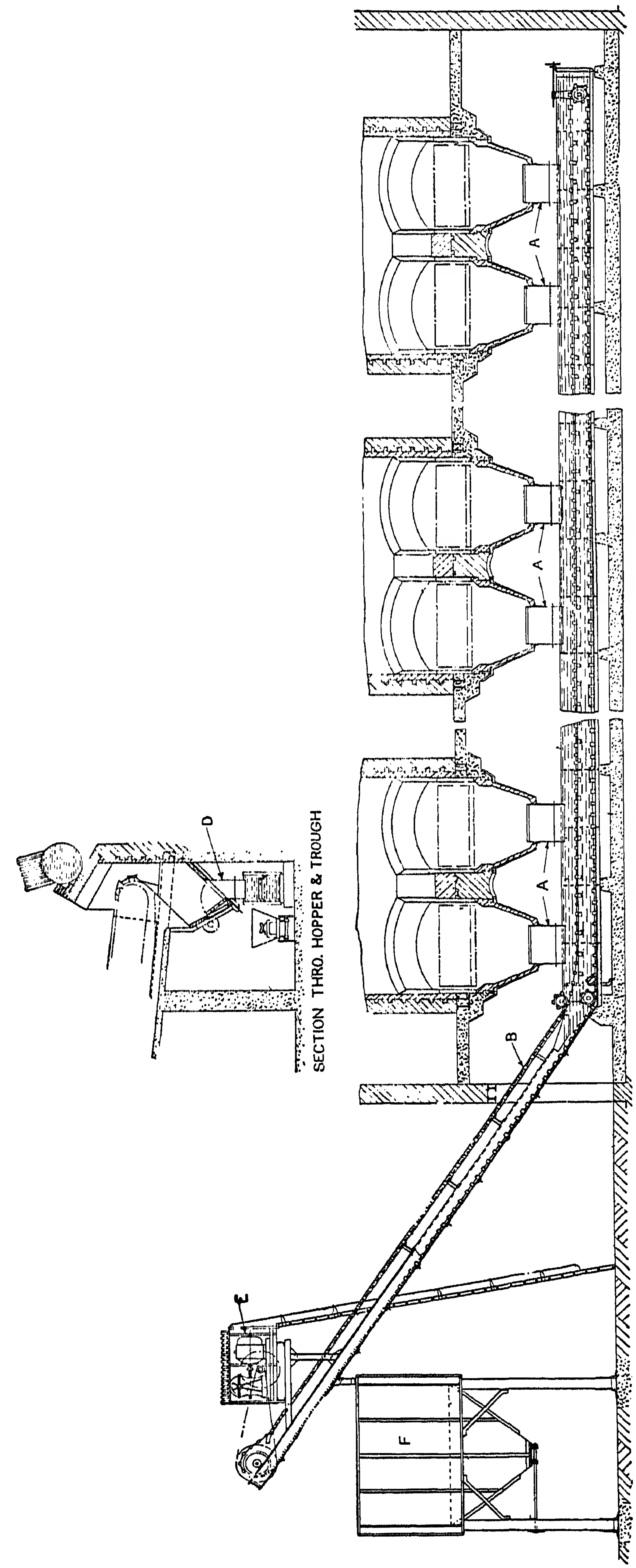

Fig. 25.—Arrangement of "Usco" Ash Conveyor

instance, in a spigot and socket water main, as the vacuum to be maintained by the joints is small, and it is an advantage to be able to quickly draw the packing from a joint for the replacement of a pipe.

The wear on a straight length of piping is greatest on the bottom, and this wear should be carefully noted from time to time, and when it reaches the permissible limit the collar joints on the pipes should be slackened where necessary, and the whole length of piping turned round on its axis, say 90°, so as to bring an unworn surface into the position of greatest wear In this way the pipe may be worn more or less uniformly all round, and it

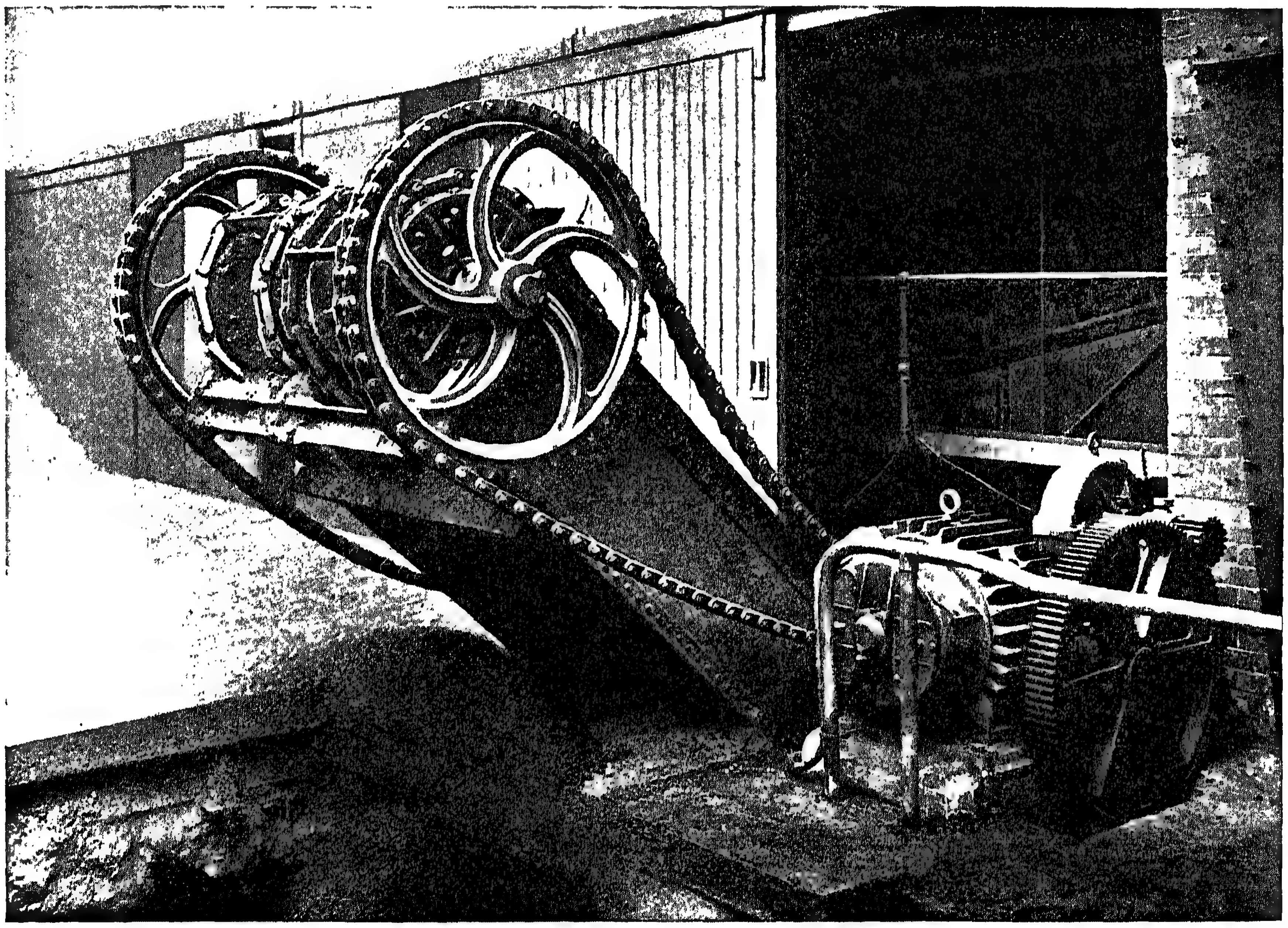

Fig. 26.—Discharge End of Water-sealed Ash Conveyor

will be appreciated that the life is thereby increased three or four times.

Water-sealed Ash Conveyor.—This system of ash removal, developed by the Underfeed Stoker Company, Ltd., consists of a scraper conveyor running in a cast-iron or concrete trough which is filled with water. The ashes from the rear end of the boiler grates are discharged by water-sealed shoots direct into the trough, whence the ashes are continuously removed and discharged by the conveyor.

Fig. 25 shows more or less diagrammatically a typical general arrangement of this special type of ash handling plant, known as the USCO Ash Conveyor. The water-sealed shoots from the boiler grates are shown at A, while the conveyor and trough are shown at B and C. The ash shoots are shown in section at D, from which it will be seen that an emergency ash door is provided for the extraction of the ashes in the event of the conveyor being out of commission. The conveyor chain is driven by motor and gearing

at E, and the ashes are discharged in a thoroughly quenched condition into an ash hopper F. The trough under the water-sealed shoots may be of cast iron or of concrete, and in either case the bottom of the trough should be provided with loose cast-iron wearing plates for the chain to run on. A continuous flow of water is maintained in the trough in order to keep the temperature of the water low enough to prevent steaming. It has been found in practice that a relatively small water-supply is all that is required to keep the trough cool.

It will be appreciated that this type of plant offers many advantages, as the perfect water seal prevents the admission of air into the combustion chambers of the boilers, which always happens in other systems in which the ashes are discharged into the atmosphere.

The operating costs are reduced to a minimum, as the plant is entirely automatic and the removal of ash continuous. No fumes or dust are given off, and as all parts of the plant are slow moving, the wear is reduced to a minimum.

In fig. 25 the ashes are discharged into an overhead ash-receiving hopper from which they are emptied periodically into railway trucks. This arrangement, however, is not always necessary, and the arrangement shown in fig. 26 is sometimes used, in which the ash conveyor discharges direct into an ash dump, from which the ashes may be loaded into trucks or on to barges by a crane and grab.

ENGINEERING CHEMISTRY

BY

W. CARRICK ANDERSON, D.Sc.

Engineering Chemistry

It is the province of the mechanical engineer to obtain command of supplies of energy and by means of suitable appliances to direct these with maximum efficiency to the performance of work. In spite of the vast development of water powers, the utilization of heat energy derived from chemical action still remains the most important source of mechanical power. Practically speaking, it may be said that the whole of this energy is obtained by promoting a chemical reaction between compounds containing carbon and hydrogen on the one hand and the oxygen present in the air on the other. These changes are exothermic, that is to say, after deduction of the energy required to break up the compounds of carbon and hydrogen employed, there is a surplus of energy set free as heat. This surplus reaches its maximum when the whole of the carbon is converted into carbon dioxide, or carbonic acid, and the whole of the hydrogen into water.

Fuels.—A material which is caused to enter into chemical action with the primary object of setting free a quantity of heat energy is a fuel. The practical fuels consist of vegetable matter or of the remains of vegetable structures metamorphosed in a greater or less degree by the action of geological forces. Such materials may be used in their raw or natural state as wood and woody by-products, peat, lignite, bituminous coal, anthracite, petroleum, and natural gas. They may also be subjected to a process of preparation or adaptation before use, and furnish the products known as charcoal, char, coke, briquettes, tar, oils, benzol, petrol, coal gas, producer and water gas. From the standpoint of the engineer who has to devise and operate appliances for the combustion of fuels, a more convenient classification is based on the state of aggregation of these bodies as solid, liquid and gaseous fuels.

SOLID FUELS

The solid fuels in use consist of wood, lignite, coal, anthracite, charcoal, partially coked coal or char and coke. Pulverized coal from its mode of combustion is more akin to a liquid fuel. Peat, on account of its bulky character, low heating power, and large content of water even when air-dried, is not an industrial fuel, although used domestically and, to a very limited extent, in distilleries. Similar disadvantages attend the use

of wood as fuel and its employment is only justified when coal is not obtainable or only at a prohibitive price. The calorific value of air-dried wood may be reckoned at 7400 B.Th.U. gross, but, since the large amount of water in its cells and formed by its combustion must be gasified at the expense of the liberated heat, this figure presents an altogether too favourable view of its value in relation to the more concentrated fuels.

Lignites or brown coals are geologically more recent than the true coals. They contain larger percentages of hydrogen and oxygen, and in consequence of the latter have generally a considerably lower heating power than coals with the same amount of ash. The water present in the air-dried fuel is generally high, and, for the reasons stated under wood although in a less degree, the low or net calorific value has generally a lower ratio to the gross than in the case of coals. When cheaply mined, lignites constitute a valuable source of heat, and they are extensively employed on the continent of Europe and elsewhere, many industries being dependent on them; but, in competition with coal, they will bear transit charges only over a limited range. They are of no importance in Great Britain, but are largely found in Western America and Canada. The following exemplify the composition of this class of fuel:

	General Composition of Organic Matter.	Lignites of Istria and Dalmatia.
	Per cent.	Per cent.
Carbon	65	63·35
Hydrogen	7	4·83
Oxygen	28	12·49
Nitrogen		1·31
Sulphur	–	8·08
Ash	—	9·94
Water		1·56
	100	101·56

Coal.—Coals consist of the altered remains of vegetable growths, generally intermixed with more or less earthy matter. The ash of coal is constituted of these foreign earthy bodies along with a small amount of mineral matter derived from the constituent substances of the plants. Three types of mineral predominate in that portion of the ash which is accidental, namely siliceous bodies resembling clay, carbonates of lime and magnesia, and iron pyrites. What may be designated the inherent mineral constituents of the coal substance are generally rich in oxide of iron. The proportions of these elements of the ash have considerable influence on the behaviour of the fuel containing it in the furnace. (See *Coal ash*, p. 36.)

The combustible matter of coal consists of a mixture of substances of complex chemical composition. Derived for the most part from leaves, spores and spore cases, and the like, and scarcely to any extent apparently from the cellulosic or woody constituents of plants, the material has, as the result of fermentative action, assumed new combinations, the nature of which is still obscure. The mother substance of a part of the organic matter of coal

probably to be found in the ligno-celluloses; another part owes its origin the albuminoid constituents of the plants that contributed to its formation. asic bodies resulting from the degradation of the latter may be conceived having entered into chemical union with acid decomposition products of e other to give highly complex compounds of carbon, hydrogen, oxygen, trogen, and sulphur; and these, along with others consisting of carbon, ydrogen and oxygen alone, now constitute the main portion of the combustible atter. At one time the mass, or a portion of it, has apparently existed in a ate of plasm or jelly, and this circumstance accounts for the fact of fundaental importance that the burning of coal, or of its derivatives char and coke, the fire is a slow and difficult matter in comparison with wood and charcoal, hich possess a tubular structure that admits a ready penetration of air. oals when destructively distilled in the furnace leave a charred mass of ells which, though not gas-tight in the absolute sense, are practically imenetrable by the air current. Combustion of this fixed carbon is thereby estricted to that portion of the outer surface of the pieces which is freely xposed to air passing through the furnace.

Classification of Coals.—It is not possible to make strict or scientific lassification of coals, since the distinguishing characteristics shade insensibly ito one another, and a single seam may contain sections that are bright and ull, yield variable proportions of gasifiable constituents, and exhibit a greater r less degree of coherence in their cokes. For use as fuels they may be rouped into (*a*) the soft coals, generally free burning and yielding more han 45 per cent of gasifiable constituents reckoned on the combustible matter; *b*) the light steam coals, which may either be free burning or cake on the fire nd yielding from 36 to 45 per cent of volatile matters; (*c*) the heavy steam oals similar to (*b*), but yielding only from 15 to 36 per cent of volatiles; and *d*) the semi-anthracites with 11 per cent to 15 per cent, and anthracite vith less than 11 per cent of volatile matters. Increase of fixed carbon is enerally accompanied by less combustibility, but not by lowered calorific alue.

Sampling of Coal.—For the testing of coal and coke the importance f careful sampling cannot be overrated. Coal is essentially an irregular nineral. Apart from the possibility of large pieces of stone and shale—soalled "dirt"—being included, the actual coal substance is not homogeneous. n the case of screened coal the portion passing through the screen generally ontains a higher ash than the round coal. In railway wagons during transit he finer portion settles to the bottom and contains more than the average sh. Two consecutive boiler trials carried out with coal from the same eap, bunker, or truck may on this account show differences in the calorific alues of the coal used amounting to as much as 10 per cent.

For large coal, sampling may be done by taking half-shovelfuls from the eap at the time of weighing each lot for use in a boiler trial, the pieces being irst broken and a proportion of large and small taken. The aggregate of samples is broken up, well mixed, and reduced by quartering. Washed small coal, washed nuts, peas, and dross are generally more regular in composition

as regards ash, but the moisture content in such materials is liable to considerable variation. A half-shovelful should be taken from each lot that is weighed and placed at once in a box with close-fitting lid. The collected samples are then spread out on a tared metal tray and accurately weighed. The tray is left in a warm place at a temperature not exceeding 120° F. for about 48 hr., when it is again weighed and the loss of moisture noted. Unwashed small coal may be sampled in the same way, with omission of the air-drying, unless it contains much moisture, when it should be treated exactly as washed coal. In all cases the reduction of the sample to manageable size is effected by the method of quartering. The aggregate samples are broken up on a metal plate to pieces not exceeding 2-in. cubes, well mixed, and piled evenly in a cone, the pieces being allowed to distribute themselves regularly on all sides. The cone is then evenly flattened out and divided by two diameters at right angles; two opposite quadrants are removed, care being taken to include their residual " fines "; the other two are rejected. The retained portion is well mixed and the quartering repeated as often as necessary with progressive reduction in the size of the pieces until only a few pounds is left. For accurate determination of the contained water it is advisable to take a portion at this stage, as in the case of many coals there is rapid loss of a portion of the contained water during fine grinding. The grinding is done in an iron mortar with further quartering until about half a pound is left, which must be reduced to the condition of fine flour. In this state it is spread out overnight in a place free from dust, and is then in the condition of air-dry sample. This is used for the analysis and for determination of heating power.

It cannot be too strongly insisted on that the utmost care is necessary in the case of nearly all coals to ensure the obtaining of a correct sample. Errors of 1 or 2 per cent are quite easily made in the case of coals containing 10 or 12 per cent of ash, a considerably higher degree of inaccuracy than is likely to occur in the analysis or in the determination of heating power if these are in skilled hands.

Proximate Analysis.—For the control of combustion it is important to ascertain the moisture and ash contents of the coal and in addition the amounts of fixed and volatile matters formed on exposure to heat. These data are supplied by the " proximate analysis ".

Moisture and Ash.—3 gm. of the finely powdered and air-dried coal is heated for 1 hr. in a suitable oven at 220° F., cooled in a desiccator, and weighed. The loss in weight is taken as moisture. For ash the dry residue from the moisture determination is carefully burned off in a muffle or over a good Bunsen flame till constant in weight.

Volatile Matter and Fixed Carbon.—1 gm. of the finely powdered and air-dried coal in a crucible with close-fitting lid is ignited for 2 min. over a Bunsen flame and thereafter, without cooling, for 3 min. over a foot blowpipe. The crucible is then cooled in a desiccator and weighed. The residue is " coke " and, after deduction of the contained ash as determined, gives the " fixed carbon ". The loss on ignition less the ascertained moisture

is the "volatile matter". These results for the air-dried sample may be corrected, if desired, to the "sample as received" by taking into account the moistures in the original and in the air-dried sample. In carrying out the coking test as described, the bottom of the crucible is set at a height of 3–6 cm. above the top of a Bunsen flame at least 18 cm. in height. This method is more reliable than the so-called American method, according to which 1 gm. of the sample is heated for 7 min. over the full flame of a Bunsen burner, the bottom of the crucible being 6 to 8 cm. above the top of the burner and the flame 20 cm. high.

Ultimate Analysis.—In the case of boiler trials where the "low" or net calorific value is required in order to determine the efficiency, it is necessary to ascertain the amount of hydrogen as well as the water present in the fuel used. For this an "ultimate" analysis of the fuel is made in which the carbon and hydrogen are burned to carbon dioxide and water respectively. The operation is performed according to the method and with the apparatus used by chemists for the general analysis of organic substances. Details will be found in laboratory textbooks on organic chemistry. In view of the general presence of sulphur compounds in commercial fuels, suitable provision must be made for the retention of sulphur dioxide formed in the combustion. The following figures give the results of actual analysis of commercial steam coals and are broadly representative of the types of fuel met with, but in view of what has already been stated regarding the composition of coals generally they should be regarded only as illustrative:

	Welsh Navigation (1).	Welsh Navigation (2).	Scotch Navigation (Lanarkshire) (1).	Scotch Navigation (Lanarkshire) (2).	Scotch Navigation (Fife).	Japan (Miike).	India (Bengal).	West Africa (Udi).
Carbon ..	88·93%	85·19%	84·42%	82·84%	81·95%	75·62%	73·62%	63·40%
Hydrogen ..	4·64	4·79	4·62	5·29	5·24	5·91	6·23	5·91
Oxygen ..	0·85	1·62	0·98	4·71	3·82	5·16	4·69	10·63
Nitrogen ..	1·42	1·42	1·58	2·00	1·97	2·54	1·73	1·21
Sulphur ..	1·30	2·07	0·94	1·28	1·29	(included with oxygen)	0·39	0·97
Ash	2·18	4·07	5·33	2·40	4·00	9·59	9·81	8·85
Moisture ..	0·68	0·84	2·13	1·48	1·73	1·18	3·53	9·03
	100·00	100·00	100·00	100·00	100·00	100·00	100·00	100·00
Volatile matter	14·39%	18·61%	18·60%	27·20%	27·32%	39·92%	37·07%	34·77%
Fixed carbon..	82·34	75·73	73·77	68·53	66·54	49·31	49·52	45·07
B.Th.U. per lb.	15,104	14,510	14,047	14,458	14,099	13,986	13,014	11,950
Specific gravity	1·336	1·324	1·350	1·303	1·316	—	1·329	1·372

Prepared Coals.—To suit the requirements of the market and at the same time to eliminate a portion of the non-combustible matter, coal as mined is frequently subjected to a process of grading with or without the accompaniment of washing. The products, in addition to round coal, are treble, double, and single nuts, peas and pearls, and dross. The tendency is for the treble and double nuts to show a reduction of ash as compared with

the average of the seam, with some concentration in the peas and pearls, and more in the dross. Unwashed drosses are sometimes compressed into briquettes with or without addition of a binding agent for use in steam raising. The following analyses exemplify these types of fuel.

	Lanarkshire Washed Single Nuts.	Dross supplied to Electricity Station (Scotland).	Peas supplied to Electricity Station (London).	Derbyshire Slack.	Eggets used on Steamer (1).	Eggets used on Steamer (2).
Carbon	66·22%	56·23%	60·38%	72·41%	74·32%	[illegible]
Hydrogen ..	4·58	4·78	4·65	4·06	3·72	[illegible]
Oxygen	6·93	7·79	8·67	9·90	5·18	[illegible]
Nitrogen ..	1·56			1·40		
Sulphur	0·83	—*	—*	1·17	*	*
Ash	6·46	16·93	14·00	3·33	15·00	[illegible]
Moisture ..	13·42	14·27	12·30	6·74	1·78	[illegible]
	100·00	100·00	100·00	100·00	100·00	100·00
Volatile matter ..	32·31%	30·65%	30·52%	-	15·00%	[illegible]
Fixed carbon ..	47·58	38·15	43·18		[illegible]	[illegible]
B.Th.U. per lb.	—	10,366	10,904	12,775	12,706	[illegible]

* Not determined.

Coal Ash.—The non-combustible matter, or ash, of coal varies greatly in amount and likewise in composition. The basis of it consists generally of silica and alumina in combination, but there are also present lime, magnesia, alkalis, and iron, sometimes in considerable quantity. Iron present as pyrites becomes converted into oxide during combustion, and carbonate of lime, sometimes found as a deposit in the cleavages of the coal, is calcined at the same time to quicklime. The composition of the ash is important as it determines the fusibility and therefore the tendency to clinker on the fire bars. With few exceptions the ashes of coals may be said to lie within the following limits of composition:

	Per cent.
Silica	25–56
Alumina	17–35
Oxide of iron	5–25
Lime	2–20
Magnesia	1–10
Sulphuric anhydride	1–3

When it is constituted by silicate of alumina with little admixture, the ash is not readily softened by heat, nor does the addition of lime alone exercise a marked influence in this direction. Silica alone melts at about 3330° F. and compounds of silica, alumina, and lime containing nearly 40 per cent of the last named have melting-points in the neighbourhood of 2730° F. and over. The oxides of iron have a much more decided action than lime in lowering the melting-point, and this is particularly the case with the lower oxide (FeO), which in moderate percentages produces a distinct tendency to clinker. Experience has been cited which shows that up to 8 per cent of

iron produces a degree of fusibility that causes little practical difficulty but that with amounts in excess of this figure troubles rapidly accumulate, and that in some cases when the percentage of iron in the ash exceeds 15 it is impossible to work the coal in an ordinary furnace. The following softening temperatures afford a classification of coal ashes in respect of the clinkering property:

1. Readily fusible.	Softening	below 2190° F.
2. Fusible.	,,	between 2190°–2460° F.
3. Difficultly fusible.	,,	,, 2460°–2730° F.
4. Very difficultly fusible.	,,	,, 2730°–3000° F.
5. Refractory.	,,	above 3000° F.

The ash of coals for use under boilers should have a melting-point over 2550° F. The percentage of ash in a fuel is without influence on its melting-point, but on the other hand the amount of fixed carbon, pointing to a larger or smaller proportion of the total combustible being available for consumption on the grate and there raising the temperature of the ash, will have some effect either towards increasing the tendency to clinker or to lessen it.

Fine Dust.—The non-combustible portion of coal is partly distributed through the mass of the fuel and partly associated with it in the form of stones and shale. Combustion of the carbonaceous matter sets free the former in a very fine state of division, so that it is easily lifted and carried away by the gases passing to the chimney. In the case of water-tube boilers with high consumption of fuel per square foot of grate area, the fine-dust problem is one requiring careful attention. The composition of this dust varies with that of the coal ash and is generally similar to it, but even when the ash is non-clinkering it is possible to have the formation of considerable masses or "nests" of semi-fluxed dust on the surface of boiler tubes towards the chimney end of the furnace, the production of which is explained by the formation of ferrous oxide on the surface of the tubes from the excess air passing through the furnace, and its subsequent union with silicates in the fine dusts that settle upon it.

Pulverized Coal.—The difficulty that attends the combustion of coal owing to its compact and impenetrable character has already been touched upon. The limitation placed on the rate of combustion owing to the latter being confined to the surface of the fuel results in a lower temperature being attainable than would otherwise be the case. Fuels high in ash are for the same reason difficult to keep alight. Considerations of this kind have led to the development of systems of burning solid fuels by reducing them first to a fine powder whereby the surface area is enormously increased, and introducing the powder into the furnace along with the air required for complete combustion. The air may be previously heated. In Britain the method has been adapted principally to the manufacture of Portland cement, but in America and elsewhere large installations for steam raising have been erected and are stated to give boiler efficiencies of 80 per cent with fuels not previously useable either with mechanical stoking or in gas producers. Bituminous

coal containing up to 30 or 40 per cent ash may be thus utilized. The moisture content with coal is generally brought down to 1 per cent by previous drying, but in the case of lignites and peat a higher percentage of water may be accepted. The standard fineness adopted is 82 to 85 per cent through a 200-mesh screen (40,000 holes per square inch) and 95 per cent through a 100-mesh screen. Three methods are in use for conveyance of the dried dust to the burners, viz. (1) screw conveyors from bins; (2) combined pulverizers and fans on the induced-draft principle; and (3) the pneumatic system, in which the pulverized fuel is carried along suspended in air and simply tapped off as and where required. The "primary" air so used is about half the total needed for complete combustion, the balance being admitted into the furnace from a secondary air main. From 20 to 40 per cent saving has been cited in comparison with solid coal by using the fuel in this finely divided state.

LIQUID FUELS

A practical division of liquid fuels would be into those which have been purified by distillation and are sufficiently volatile to be used as vapours in the cylinders of internal-combustion engines, and those whose volatility is so low as to require breaking up mechanically into minute liquid globules for rapid combustion. The first group contains the petrols, benzol, alcohol, alcohol-benzol mixtures, white spirit, paraffin and kerosene oils; the second the crude fuel oils, obtained for the most part by "topping" or distilling off the more volatile portions of the natural petroleums found in various parts of the world. These distillates yield petrol, kerosene, and other grades of refined mineral oil. Benzol is similarly obtained from the light oil of coal tar and is sold as 90 per cent and 50 per cent benzol, terms which imply that in the one case 90 per cent and in the other 50 per cent distils over at 212° F. from a glass retort, the thermometer bulb being immersed in the liquid. 90 per cent benzol contains about 75 per cent of actual benzene, having the composition C 92·3 per cent, H 7·7 per cent, and 24 per cent of toluene (C 91·3 per cent, H 8·7 per cent) with 1 per cent of higher hydrocarbons. Pure alcohol, free from water, contains C 52·17 per cent, H 13·04 per cent, and O 34·79 per cent. The petrols consist almost entirely of hydrocarbons, but the products from different oil-fields show differences in constitution, a paraffinoid structure prevailing in American oils, naphthenic in Russian, and benzenoid in that from Borneo. The calorific value of petrol is about 19,500 B.Th.U.

Fuel Oils.—Heavy fuel oils suitable for combustion in furnaces are got from Borneo, Burmah, Persia, California, Texas, Mexico, Trinidad, Russia, Galicia, Roumania, and elsewhere. Scottish shale and blast-furnace oils have also been used. As the fuel oils have generally been subjected to topping, their analyses afford no information regarding the composition of the raw oil from the various fields, and the titles given below must be taken as indicating only the alleged source of supplies offered in the market. The

last column gives the average composition of four typical oils as analysed by Brame.

ANALYSES OF LIQUID FUELS

	Borneo.	Burmah.	Texas.	Mexico.	Russia.	Scotch Shale.	Scotch Blast-furnace.	Average of Four Typical Oils.
Carbon ..	87·80%	86·40%	83·84%	85·85%	84·94%	83·77%	82·30%	84·70%
Hydrogen	10·78	12·10	12·48	10·82	13·96	13·25	10·11	11·50
Oxygen, &c.	1·24	1·50	3·68	3·30	1·25	2·98	7·59	3·49
Sulphur ..	—	—	—	—	—	—	—	0·35

The suitability and value of a particular oil for fuel purposes must be determined by its calorific value (net), viscosity, flash-point, and purity in respect of water, sand, and sulphur independently of its source.

Calorific Value.—The average calorific value of thirteen varieties of fuel oil as determined by Brame was 19,200 B.Th.U. (gross), but this is probably rather above the general figure for market oils. The value should not fall below 18,000 B.Th.U.

Viscosity.—Viscosity varies within wide limits. It is lowest in the case of shale oils, while Mexican oils are sometimes very viscous.

Flashpoint and Specific Gravity.—For Admiralty use the flash-point should be above 175° F., and in the case of low viscosity oils, such as shale, should reach at least 200° F. In the case of the mercantile marine a flash-point of 150° F. and over is suitable. The specific gravities of liquid fuels range from 0·875 for shale oil to 0·96 in the case of the heavier earth oils and 0·98 for blast-furnace oil. When comparison is made between mineral oils of similar constitution, a relation is established between specific gravity and volatility, the specific gravity showing a progressive increase as the volatility diminishes till the solid oils are reached. The fact, however, that the oils from different fields are differently constituted, and the presence in them of variable amounts of oxygen and sulphur compounds, make it impossible to extend the relationship to oils derived from different sources.

Combustion of Oils.—For the complete combustion of oils it is necessary to ensure their efficient dispersal through the air required for combustion. In the case of the more volatile oils this is done by converting into vapour in a suitable carburettor. The heavy oils used for combustion under boilers are atomized either by means of a steam jet or by air or by the direct breaking-up into spray of a jet of oil caused to issue under pressure. The difficulty is in all cases to ensure regular and smokeless combustion. Steam has the drawback of rendering latent a part of the heat which is carried away in the gases of combustion, and steam atomizers have been characterized further as not responding as well as air and pressure systems to any forcing of the boilers. The following description of the Wallsend-Howden pressure system which has been widely adopted on boilers for marine and land use will serve to illustrate the principle of mixing the air-supply with the oil.

On board ship the oil is carried in special tanks, or in the double bottom, and when viscous oil is used an arrangement of steam-heating coils is placed in these to liquify the oil sufficiently to allow of a service pump delivering it from them to settling tanks in the engine-room. In the settling tanks the water is drained off, and the oil pressure pump then draws the oil through suction strainers and delivers it to the oil heater, where it is heated by steam coils to a temperature of 150° to 270° F., according to the class of oil used. The oil is delivered to the burners at a pressure of 60 to 80 lb., and the burners are designed to atomize the oil sufficiently and give a flame directed at the proper angle. On the furnace fronts, to which the burners are attached, air directors are fitted, the burners passing through the centre of these. The directors are fitted with angled vanes which give the air a rotary movement, and the latter as it issues becomes intimately mixed with the fine oil spray from the burners, so that complete combustion takes place with production of a clear white flame free from smoke.

GASEOUS FUELS

With the exception of natural gas, which escapes from the earth in quantity only in a limited number of localities, the gases used as fuel are prepared in appliances primarily designed for that purpose, or are got as by-products. To the first category belong coal gas, water gas, producer gas and Mond gas; to the second coke-oven gas, which is in composition similar to coal gas, and blast-furnace gas, which is somewhat akin to producer gas in character but carries in it a larger amount of fine dust.

Coal Gas.—Coal when directly distilled to coke yields about 10,000 to 12,000 c. ft. of gas per ton, the composition of which varies within comparatively narrow limits according to the character of the coal used and the nature of the retorts, horizontal or vertical, in which it is treated. Greater departures from the normal result from admixture with more or less water gas, and it is not safe to-day to assume that lighting gas is identical with coal gas in the chemical sense.

Producer Gas.—When the fixed carbon of coal is burned in a limited supply of air, carbon monoxide is produced mixed with the nitrogen of the atmosphere and a little derived from the coke. This mixture of carbon monoxide and nitrogen approximately in the proportions:

	Per cent.
Carbon monoxide	34·7
Nitrogen	65·3

is producer gas. If, however, the producer be fed with raw coal instead of with coke, as is generally the case, the resulting gas will carry with it the coal gas formed from the destructive distillation of the coal and also the gases formed from the destruction of the tar. Producer gas so formed is somewhat richer than that made from coke, and has a calorific value of about 135 to 140 B.Th.U. per cubic foot.

The combustion of fixed carbon to form carbon monoxide is an exothermic reaction, and carbon treated thus evolves in the producer about 30 per cent of the heat energy it would yield if burned completely to carbon dioxide. A large part of this heat is lost in such circumstances through radiation. Hence has arisen the practice of introducing into the producer with the air blast a certain amount of steam. Steam, when brought into contact with incandescent carbon, is decomposed with formation of hydrogen and carbon monoxide. This is a heat-absorbing, or endothermic, reaction, and by careful combination with the other can be made to effect a considerable reduction of the heat loss occurring in the producer. It should be noted that this is the only benefit resulting from the use of steam (apart from any increase in by-products). It does not effect a larger yield of heat actually from the fuel, but it lessens the heat loss in the producer, utilizing a part in the formation of hydrogen which by its subsequent combustion will set free an equal amount to that absorbed from the producer.

Water Gas.—When the exothermic reaction $2C + O_2 = 2CO$ is employed to raise the temperature of the coke in a producer to incandescence, and the air blast is then shut off and steam passed in alone, the endothermic reaction $H_2O + C = H_2 + CO$ takes place with, in consequence, a continuous fall in the temperature of the coke mass. As the temperature falls, the gases become more and more contaminated with carbon dioxide with concomitant lowering of heating value, so that the steam-supply must be turned off and the temperature of the coke again raised by the introduction of air. The mixture of hydrogen and carbon monoxide got in this way without admixture of producer gas is called water gas.

Mond Gas.—When a gas producer is fed with air and steam and the steam is kept in large excess—about 2½ tons per ton of coal—a large yield of valuable by-products is obtained from the coal along with a gas of about 145 B.Th.U. per cubic foot. As a result of the cooling produced by the steam the gas contains a high proportion of carbon dioxide, but the efficiency of the process is high. The gaseous mixture obtained in this way is known as Mond gas.

Examples of these different types of gaseous fuel are given below:

	Coal Gas (1).	Coal Gas (2).	Producer Gas.	Water Gas.	Steam-fed Producer Gas.	Mond Gas.	Blast-furnace Gas (Coke Fed).	Blast-furnace Gas (Coal Fed).
Hydrogen ..	54·0%	47·1%	4·4%	48·4%	12·11%	27·20%	2·7%	6·8%
Methane ..	34·0	36·0	—	0·5	3·43	1·80	0·2	3·0
Ethylene.. ..	3·0	4·3	—	—	—	0·40	—	—
Benzene	1·0	0·5	—	—	—	—	—	—
Carbon monoxide	6·0	8·0	25·6	43·6	22·32	11·00	28·6	27·0
Carbon dioxide ..	—	1·6	4·3	3·5	6·18	17·10	11·4	7·2
Nitrogen.. ..	2·0	2·5	65·7	4·0	55·96	42·50	57·1	56·0
	100·0	100·0	100·0	100·0	100·00	100·00	100·0	100·0

Gas Analyses.—Perhaps the most valuable property possessed by the gaseous fuels is that of diffusion, which admits of their combustion being effected under very close control and with only a small excess of air over that theoretically required. To enable the user to take full advantage of this, he must know the exact composition of the gas mixture in order that the quantity of air admitted to the furnace may be adjusted to what is necessary for complete combustion. As the highest degree of economy in working will be reached when the whole of the carbon and hydrogen in the gases has been burned to carbon dioxide and water, and when this has been effected with a minimum surplus of oxygen, it follows that an examination of the products of combustion passing to the chimney will afford confirmation of its attainment:

1. By the absence of unburned gases such as hydrogen and carbon monoxide, and
2. By the presence of only a moderate amount of free oxygen, indicating excess air.

As the products of combustion derived from liquid and from solid fuels are in the main identical with those generated from the gaseous fuels, the same methods of analysis are applicable to the waste gases of all three.

Analysis of Flue Gases.—If the general behaviour of the fuel in use is understood, it is frequently sufficient to make a determination of the carbon dioxide and the oxygen in the flue gases and to infer from the figures their condition as regards carbon monoxide and other combustibles. Where more exact knowledge of the latter is required, reference is made to what is said below on the general methods of gas analysis. However little may be deemed necessary in the way of analysis, it is all-important to make certain that the sample worked on is representative of what is passing through the flue, and that samples are taken in sufficient number at short intervals to eliminate the variations in composition of the gases at different stages of the combustion process. In mechanically-stoked furnaces these are less extreme than with hand firing. The samples are withdrawn from the flue or chimney through a glass or porcelain tube into a containing vessel which may be either a separate tube of glass closed by glass stopcocks or by pinchcocks, or may form part of the apparatus used for analysis. If the flue gases are under pressure the tubes are readily filled; if not, an aspirator may be used. A rubber aspirator is sometimes supplied with the analytical apparatus. Care must be taken to sweep out air from tubes and connections by means of the flue gases before taking the sample.

Orsat Apparatus.—What is known as the Orsat apparatus in one or other of its modifications is used for the analysis. Fig. 1 shows that devised by Lunge. It consists of a graduated measuring tube surrounded by a water jacket and connected at the foot to a movable bottle. The upper end connects to a series of glass pipettes closed by glass stoppers. Beyond these is placed a small horizontal tube containing a thread of palladiumized asbestos capable of being warmed by a spirit lamp. This

is designed to oxidize small quantities of hydrogen should their determination be called for. At the extreme end, where the gases enter the apparatus, there is arranged a filter tube with pad of cotton-wool to remove particles of smoke and soot from the sample before measurement. For a simple determination of carbon dioxide and oxygen, No. 1 pipette is charged with a solution of caustic potash, made by dissolving 1 part solid caustic potash in 2 of water; this serves to absorb the carbon dioxide. No. 2 pipette is filled with an alkaline solution of pyrogallic acid made by dissolving 20 gm. of pyrogallic acid in 100 c. c. of water and adding 350 c. c. of a strong solution of caustic potash (1 to 2). The measuring tube is filled with water, or with a solution of glycerine, by opening the outlet tap at the far end and raising the bottle. The tap is then turned to connect with the filter tube, and some of the sample is drawn in by lowering the pressure bottle. This is then ejected once or twice to sweep out air from the connections, and finally a quantity is drawn in and measured after adjustment of the water level. About 100 c. c. is used for the analysis. The gases are transferred in turn to the potash pipette, where contact is maintained till no further reduction in volume takes place, and to the pyrogallate, careful measurements of the volume absorbed being made in each case. The readings are taken after adjustment of the level as at the start, and the shrinkages in the two pipettes, reckoned per 100 c. c. of sample, give the percentages by volume of carbon dioxide and oxygen respectively.

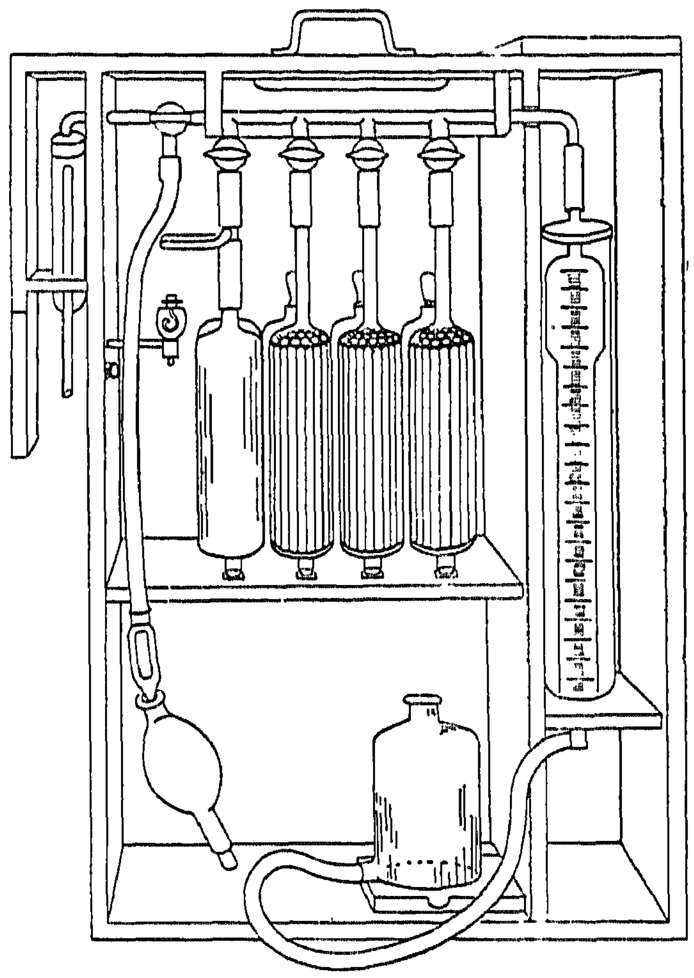

Fig. 1.— Orsat Apparatus

If it is desired to ascertain the amount of carbon monoxide present in flue gases, the Orsat apparatus may be used for this determination also. The third absorption pipette shown in the figure is filled with an ammoniacal solution of cuprous chloride, and the gas remaining after complete treatment with the liquids of the first and second is brought in turn into contact with this, till no further shrinkage in volume is observed on returning to the measuring cylinder. The final difference in volume reckoned per 100 c. c. gives the percentage of carbon monoxide. Except in cases where a considerable shortage of air for combustion has prevailed in the furnace, the remainder of the sample of gas will consist of nitrogen, the volume of which will be got by deducting the percentages of the constituents as determined, from 100. Where combustion has been incomplete, however, the possibility of hydrogen and hydrogen-containing gases being present must be taken into account.

The accurate analysis of gaseous fuels and other complex mixtures of gases is an operation demanding not only specialized apparatus but a con-

siderable degree of manipulative skill, and can be undertaken only by a trained chemist. The principles underlying it, however, are those exemplified in the use of the Orsat as described above, more elaborate apparatus being used to obtain a higher degree of accuracy, and the samples collected and measured over mercury for the same reason. Carbon dioxide, oxygen, and carbon monoxide are absorbed in the order named and by the use of the reagents already described. Ethylene if present may be absorbed by bromine. A portion of the residual gas is then taken, mixed with an excess of air or oxygen, and exploded by an electric spark while contained over mercury in a stout glass vessel known as an explosion pipette. The hydrogen is burned into water, the hydrocarbons into carbon dioxide and water. The volume of the carbon dioxide produced is equal to the amount of methane, and from the total shrinkage resulting on explosion the hydrogen is obtainable by calculation.

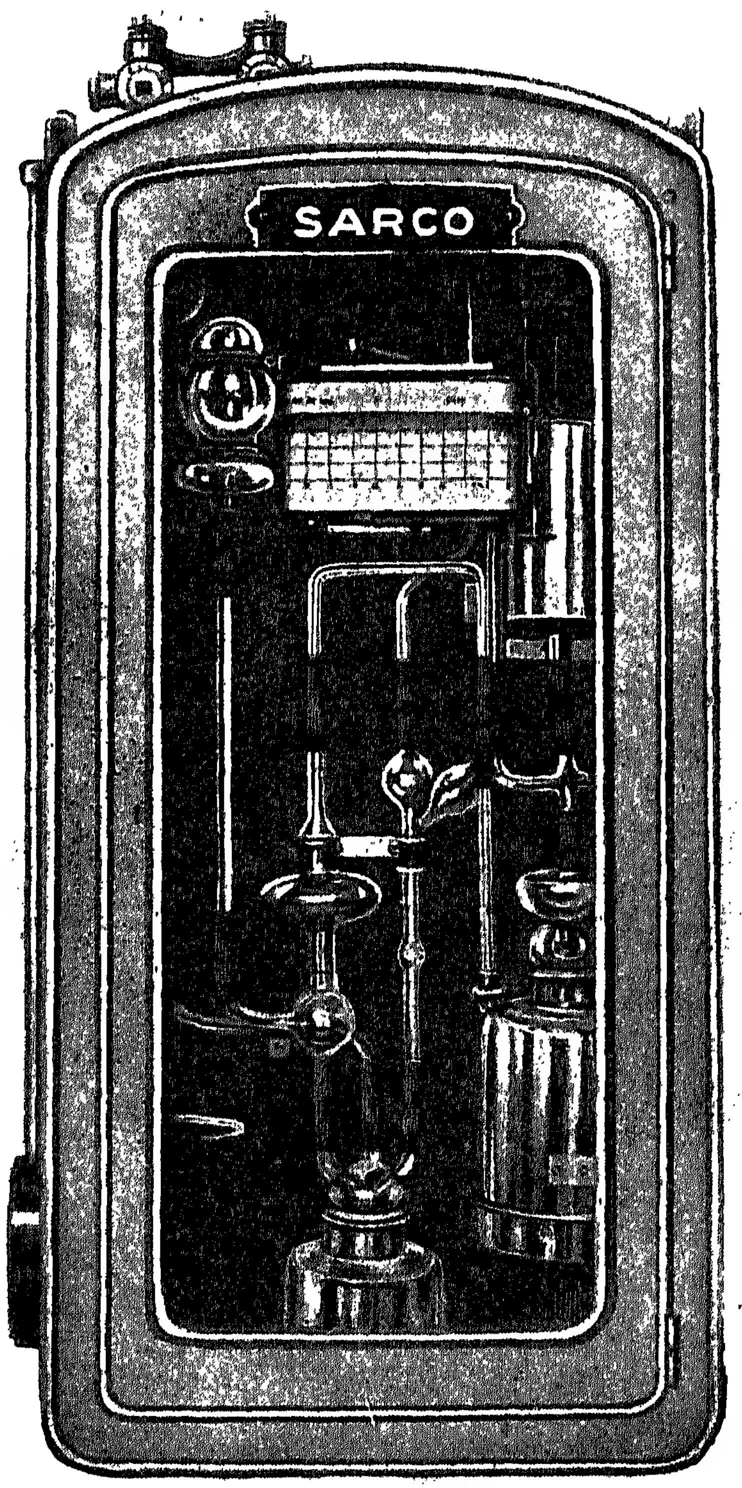

Fig. 2.—CO_2 Recorder

Carbon Dioxide Recorders.—The absorption of carbon dioxide by means of caustic potash, which is used for its determination in the ordinary methods of gas analysis, is rapid, and by the employment of mechanical arrangements such absorptions can be carried on successively and without supervision, the percentages of carbon dioxide in the gas mixture being recorded on a chart. CO_2 recorders of this kind now form part of the necessary equipment of a modern installation for the economical production of heat by combustion. A single recorder may be arranged to operate on a battery of boilers, but a more satisfactory practice is to have an instrument recording the working of each boiler separately. These may be erected singly or in a group occupying one case, and, beyond the periodical refilling of the caustic potash holder and attention to the changing of the records, they require little looking after. From a study of the charts it is possible to judge the efficiency of the furnace working from hour to hour, to adjust the draft to what is necessary for perfect combustion, avoiding too much excess air, to fix the best thickness of fire, and to detect the presence

of leakages and faults in the brickwork setting—in other words to know how far the furnace approximates to its task of converting the carbon completely into carbon dioxide without dissipating its heat on excess air.

In the " Sarco " recorder, fig. 2, supplied by the Sarco Engineering & Trading Co., Ltd., connection is made with the flue by a 1-in. pipe from which a $\frac{3}{4}$-in. tube leads to the instrument, and to ensure rapid movement of the gases a return tube connects the recorder with the base of the chimney, or other convenient outlet. Analyses are completed and the results recorded at the rate of 20 to 30 per hour with an accuracy well within $\frac{1}{2}$ per cent of the carbon dioxide actually present in the gas.

The S. A. W. patent recorder (fig. 3), manufactured by Alexander Wright & Co., Ltd., of Westminster, consists of a gas-extracting chamber into which a large sample of gas is drawn from the flues by a water injector. From this a smaller measured sample is removed and delivered to the potash container, and a small gasholder with rising bell carrying a graduated scale on top receives the gas, after removal of the carbon dioxide, and records its volume. This is simultaneously indicated on a rotating chart for permanent reference, and determinations are completed in about three minutes.

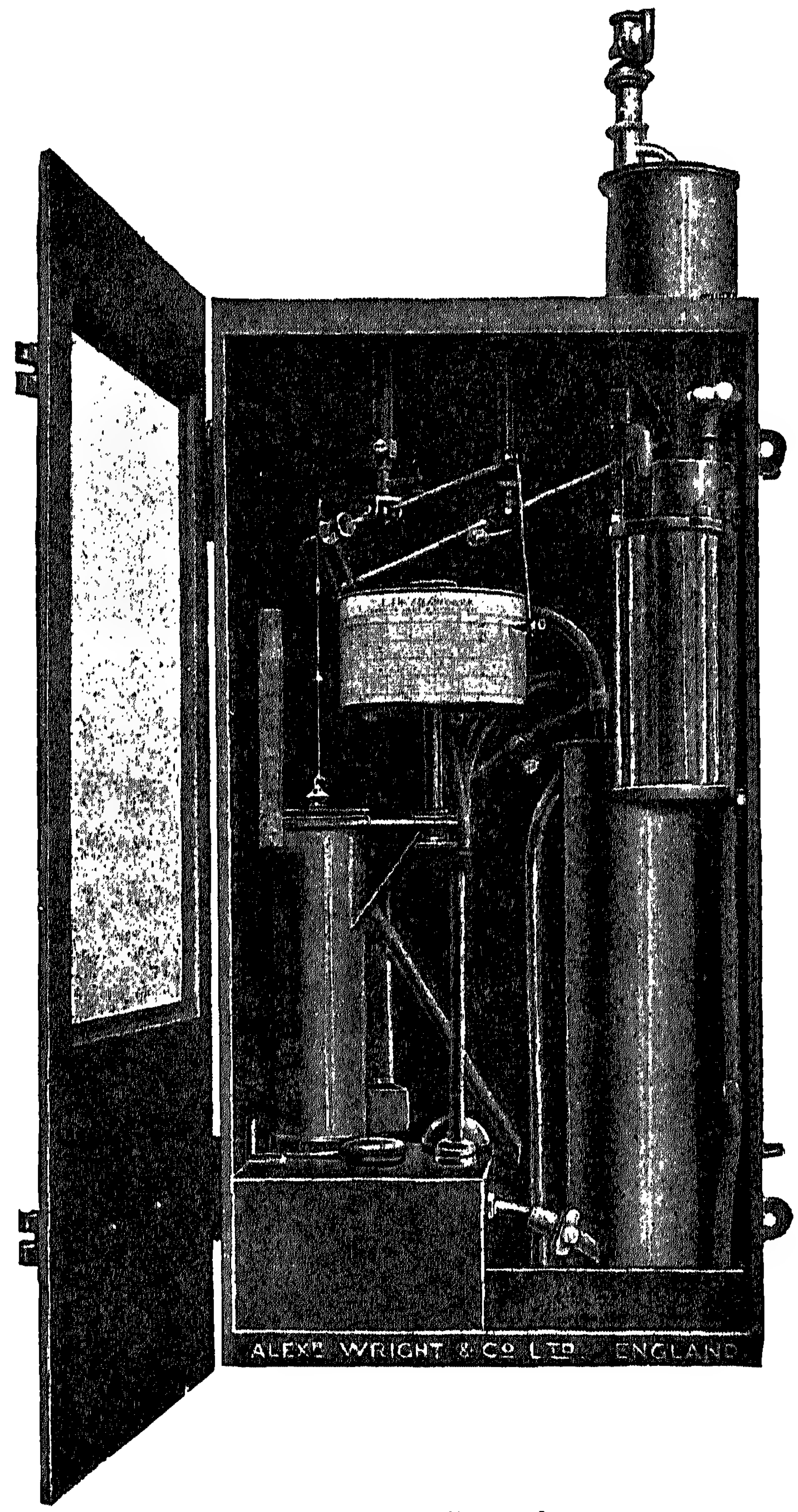

Fig. 3.—CO_2 Recorder

" Arkon " combustion recorders are placed on the market by Walker, Crosweller, & Co., London, in several forms. Model F is water-driven, but, where there is difficulty in arranging water-supply or outlet, model G may be substituted in which a $\frac{1}{3}$-h.p. electric motor is employed to draw the gas samples to the recorder and to pass them through the analysis. This type is found suitable for power stations and engine-rooms where wiring is easy but water service sometimes difficult. Fig. 4 shows the method of working of the water-driven model. Water falling into tank 10 and passing out by 13 maintains a constant level. The tube 11 is perforated inside the tank 10 so that water enters it and, falling down, draws the sample through gas burette 1, which is connected with the flue. A second stream of water falls through

tube 8 into vessel 8, driving out the air in the latter and forcing the liquid in 2 up into the gas burette 1. The left- and right-hand branches of the latter are thereby closed, and the gas sample (100 c. c.) contained in it is driven over into the potash vessel 3, where the CO_2 is absorbed. The unabsorbed portion presses an equal volume of solution up the centre tube till it lifts the glass float 7 and actuates the pen gear. Finally the water which has been rising

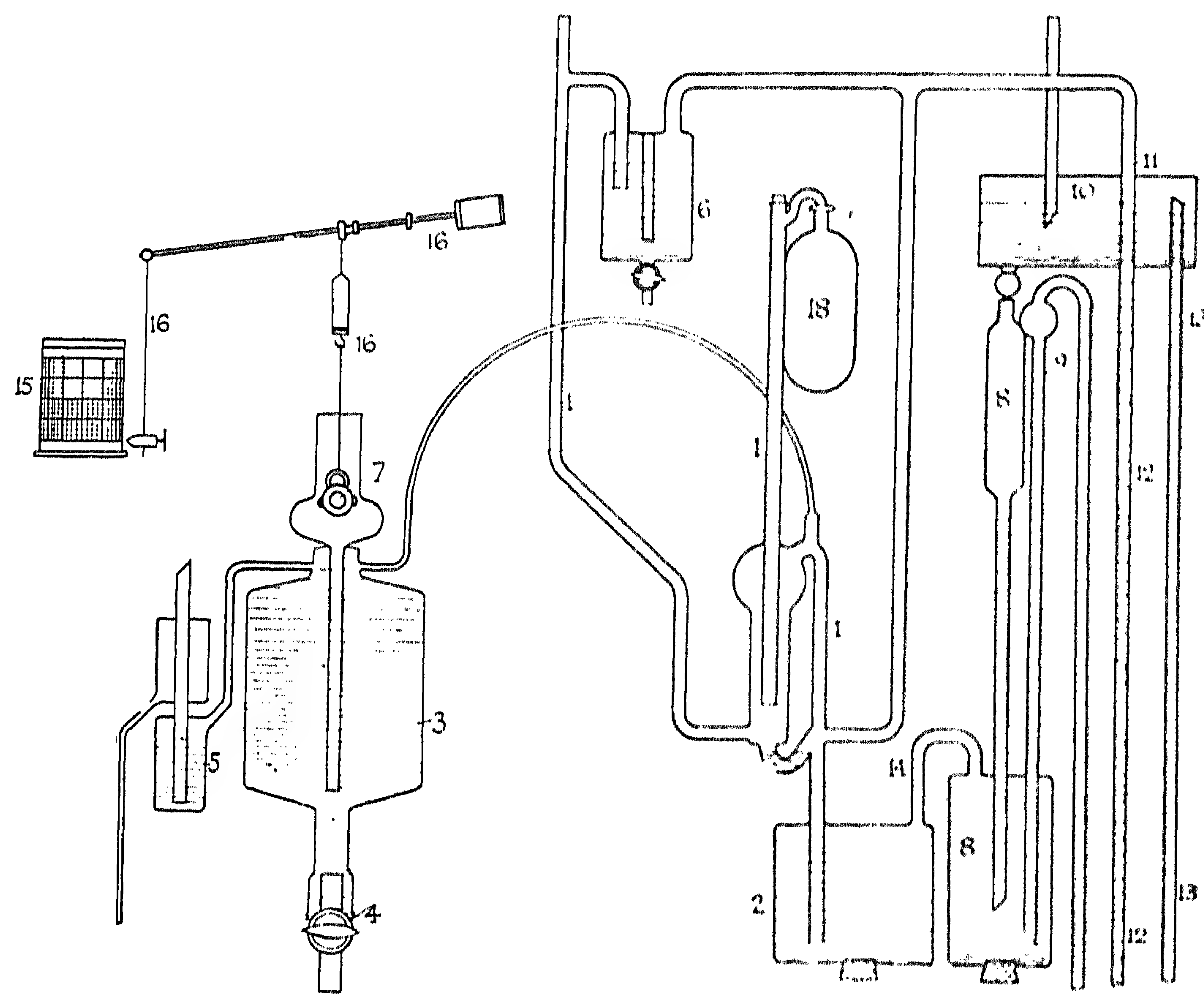

Fig. 4.—Arkon CO_2 Recorder (diagrammatic)

in 9 siphons over and discharges the water from 8, allowing the liquids in 2 and 3 to return to their normal positions. To eliminate "lag" the glycerine seal vessel 6 is provided so that gas can short-circuit across to 11 when the left and right branches of 1 are sealed. Vessel 5 maintains an exact volume of potash in the absorption vessel at all times. When the liquids are falling in the gas burette at the end of a stroke, the outer level is higher than the inner because of the gas in the central tube being sealed at atmospheric pressure, while the gas in the outer is at lower pressure owing to absorption of the CO_2. The burette is graduated at this part, and the scale provided enables the CO_2 percentage to be read independently of the chart record and supplies a check upon its accuracy.

BOILER FEED-WATER

In the matter of economical production of steam it is not sufficient to obtain a supply of heat in the furnace at minimum cost by careful selection of fuel and its complete combustion under properly regulated conditions of air-supply. It is necessary, further, that this heat should be transferred to the boiler at the highest possible efficiency. In this connection regard must be had to the quality of the water used for supplying the boiler, since this, if unsatisfactory, may add to the cost of steam-raising, not only directly by producing within the boiler incrustations that hinder the passage of heat from the furnace, but indirectly through the added expense of removing these deposits from time to time, as well as through deterioration resulting from corrosion of plates and fittings.

Impurities in Water.—Water that is chemically pure, that is to say containing neither matter in suspension nor in solution, is best suited for the work of steam-raising; but this ideal is not easily attained in practice, and it is important to notice that the nearest approach to it is not always a good second best. When impurities are present, their nature as well as their quantity must be taken into account. Oil is not uncommon as a contamination of water from surface condensers, and when passed into the boiler has been known to produce damage out of proportion to its actual amount. Freshly fallen rain-water, if collected in a clean tank, makes excellent boiler feed, but left in contact with decaying vegetation for a time may develop an actively corrosive character from the presence of organic and carbonic acids. Such a water may be more harmful than one containing many grains of saline matter and might be improved by adding to the impurities already in it. In addition to air and carbonic acid, waters that have been in contact with rocks and soil contain in suspension and in solution solid matter derived from these. The suspended matter may be got rid of by settling, or, if necessary, by filtration. In most practical water-supplies the quantity of the suspended matter is small in comparison with what may be present in solution, and its removal presents less difficulty. The solid matters dissolved in natural waters consist usually of organic matter, silica, iron, and alumina in small quantities, while carbonate and sulphate of lime, carbonate, sulphate, and chloride of magnesia, and chloride of sodium may be present in amounts varying from traces up to many grains per gallon. The carbonates of lime and magnesia are not soluble to any great extent in pure water ($CaCO_3$ $1\frac{1}{2}$ gr. and $MgCO_3$ $8\frac{1}{2}$ gr. per gallon at 60° F.), but are dissolved in the form of bicarbonates, $CaCO_3 \cdot H_2CO_3$ and $MgCO_3 \cdot H_2CO_3$, by the carbonic acid already in solution in the water. Natural waters are sometimes grouped according to their origin as rain, river, shallow well, deep well, spring, and mine water and finally sea-water, but such a classification has little practical value for the engineer, since, if the extreme members be excluded, the other categories show wide differences—the result of varying composition in rocks and soils through which the water has passed, and in some cases of local

circumstances such as contamination with industrial by-products. For the purpose of boiler feed the important points to be decided in connection with a natural water are the presence or absence of corrosive substances and the extent to which on evaporation it gives rise to hard incrustations, the so-called boiler scale. The probable character of these deposits is also a matter of interest in the choice of a water-supply for steam-raising.

ANALYSES OF NATURAL WATERS IN GRAINS PER GALLON

	Glasgow.	London.	Stoke-on-Trent.	Lincoln.	Hawick.	Colliery Feed-water.	Colliery Feed-water.
Dissolved solids ..	2·94	5·11	4·9	27·9	24·3	33·9	77·8
Mineral matter ..	1·75	3·78	4·7	22·0	22·0	31·3	76·9
Organic matter ..	1·19	1·33	0·2	5·9	2·3	2·6	0·9
Chlorine	0·65	0·40	0·9	2·5	1·1	1·5	1·9
Temporary hardness	0·44	2·45	2·0	8·1	11·0	—	29·4*
Permanent hardness	0·18	0·53	0·7	8·7	6·7	—	—
Acidity as H_2SO_4 ..	—	—	—	—	—	2·9	—

* Alkalinity: the water contains sulphate and carbonate of soda.
Sea-water contains about 2300 gr. of solid matter per gallon.

Hardness.—The bicarbonates of lime and magnesia render water alkaline to indicators such as methyl orange or alizarin paste, and more or less mineral acid is required to make waters which contain these bodies once again neutral. If the amount of acid required per gallon to effect neutrality in the water be expressed in terms of grains of carbonate of lime needed to combine with it, the figure obtained is called the temporary hardness. The temporary hardness is thus the alkalinity of the water expressed in grains of carbonate of lime per gallon. One degree of hardness is the equivalent of one grain of carbonate of lime per gallon. The bicarbonates of lime and magnesia are unstable in the heat and separate the respective carbonates when the water is boiled, and these, being much less soluble, are almost entirely precipitated. Hence the alkalinity or temporary hardness of a water as determined represents approximately the lime and magnesia thrown out as carbonates on heating the water. Deep-well and pit waters sometimes contain alkalinity due to carbonate of soda and not removed on boiling the water.

Water that contains lime and magnesia in the more stable forms of sulphates, chlorides, and nitrates does not part with these on boiling. On the other hand, if alkali be added to such a water, the bases forming these salts are precipitated, more or less alkali being used up in the process, and the amount so used up can be determined by an acid test. In this way it is possible to determine the alkali-destroying power of these substances in a gallon of the water and to express this in terms of their equivalent of carbonate of lime. The figure so obtained is the permanent hardness. The permanent hardness

is expressed in degrees per gallon, one degree being, as with the temporary hardness, the equivalent of one grain carbonate of lime per gallon.

A hard water is one that contains considerable quantities of temporary or permanent hardness per gallon; a soft water is relatively, or entirely, free from substances causing hardness. No definite line can be drawn between the two, but a water containing more than the equivalent of 7 or 8 gr. carbonate of lime per gallon would generally be accounted hard. In the preceding table the first three are soft, the others are of varying degrees of hardness.

Of the substances contributing to the permanent hardness, calcium sulphate is generally the most abundant, and the behaviour of this substance on heating requires consideration. At 93° F. it is soluble in pure water to the extent of 148 gr. per gallon, but the solubility is less at higher temperatures. At 212° F. it is about 113 gr. per gallon, while in a boiler working at 200 lb. per square inch, where the water is at 388° F., it is soluble only to the extent of 11 gr. per gallon.

Boiler Scale.—When a natural water is allowed to enter a boiler under steam, without previous heating, solid matter may separate from it in the following ways:

1. By decomposition—calcium and magnesium bicarbonates being broken up and carbonate of lime and carbonate of magnesia thrown out of solution.
2. By lessened solubility—calcium sulphate in excess of 10 gr. per gallon being thrown out at high temperatures.
3. By concentration through evaporation—residual quantities of the various solids in solution being thrown out as the solution reaches the saturation point for each one.

The deposits formed in this way are of very different kinds. Where they consist almost exclusively of carbonates they are soft and even sludge-like, but where calcium sulphate is present in quantity, as is generally the case, the deposit is crystalline in character and often of rock-like hardness. The state of aggregation of such boiler scales is influenced not only by the composition but by the rate of deposition as well. In addition to the constituents already named, they contain smaller quantities of oxide of iron, alumina, and silica, with sometimes magnesium chloride, and, in the case of sea-water, a large amount of sodium chloride. Examples of scales from a variety of sources are shown in the table on the following page.

Corrosive Waters.—As a general rule scale-forming waters are not actively corrosive, but where carbonates are present in a scale only in very small quantity, corrosion or pitting of the plates is sometimes found underneath the crust. This appears to be due in some cases to interaction between the iron of the plate and the oxygen present in the sulphate of lime, followed by solution of the resulting oxide of iron in the saline mixture.

Waters which contain magnesium chloride even in small quantity are generally suspected of being corrosive when used in boilers, in as much as, when magnesium chloride is heated with water, a reaction occurs in which

	Water-tube Boiler, Silvertown.	Water-tube Boiler, London, E.	Water-tube Boiler, Bermondsey.	Steamer Water-tube Boiler.	Steamer Water-tube Boiler.	Oily Scale from Water-tube Boiler.
Moisture	0·65%	1·42%	0·70%	1·15%	0·36%	—
Combined water and organic matter ..	—	5·02	5·70	—	—	4·82%
Silica	5·56	6·47	5·10	0·84	0·29	2·39
Oxide of iron and alumina	2·16	8·10	7·08	28·25	12·90	9·63
Lime (CaO)	47·43	39·17	37·46	29·24	34·19	34·72
Magnesia (MgO) ..	4·29	5·59	1·60	3·20	2·21	1·11
Sulphuric anhydride ..	17·02	14·68	2·67	36·58	46·65	0·21
Chlorine	—	—	—	0·72	0·30	—
Carbon dioxide	22·92	19·55	28·09	0·02	3·17	27·16
	100·03	100·00	88·40	100·00	100·07	80·04
Oil	—	—	11·60	—	—	19·96
			100·00			100·00

hydrochloric acid is set free. Ordinarily the ill effects that might be expected to result from this reaction are prevented largely or altogether by the presence of carbonate of lime and carbonate of magnesia which neutralize the acid. On the other hand there is reason to believe that, in conjunction with oxygen in the feed-water, magnesium chloride is a markedly corrosive agent, and very rapid destruction of plates and fittings of boilers at sea has been noticed which appeared traceable to their joint action. Very soft waters, if containing carbonic acid and oxygen in quantity, manifest corrosive action which shows itself particularly about the water-level. A small addition of lime shell in such cases is efficacious. Where canal or river water in industrial areas is used for feed, the possibility of discharges of acid liquors into the supply must be kept in view, and even collected rain-water in such neighbourhoods is liable to show traces of sulphuric acid derived from the atmosphere.

Purification of Feed-water.—If chemically pure water were obtainable in quantity this would be the ideal for steam-raising. In practice the nearest approach to it is got by taking the condensed steam from surface condensers and returning it to the boilers. Certain risks attend this operation and must be guarded against, notably the contamination of the condensate with oil and the presence in it of excessive quantities of carbonic acid and air. The admission of the water used for cooling must also be prevented as far as possible, more especially if this is sea or dirty river water. For the latter purpose salinity indicators are used—a specialized form of hydrometer either of glass or metal—or, for more accurate observation, a standard solution of silver nitrate with chromate of potash indicator. To make up the unavoidable shortage it is always necessary to supplement the boiler supply from outside, and for this recourse must be had to a natural water. If such is available containing only a few grains of hardness per gallon it may be used directly, but if only a hard water is to be had, this should be treated for removal of the hardness before admission to the boiler. The extended use of water-tube boilers working at high pressures in present-day practice involves a corresponding employment of artificially purified water.

The method followed almost universally, on the ground of cheapness and adaptability, consists in mixing the water with lime-water and carbonate of soda in quantities previously ascertained by experimental tests. Pfeiffer gives the following equations as indicating the nature of the reactions:

$$
\begin{array}{llllll}
(1) & CaH_2(CO_3)_2 & + CaO & = 2CaCO_3 & + H_2O. & \\
(2a) & MgH_2(CO_3)_2 & + CaO & = MgCO_3 & + CaCO_3 & + H_2O. \\
(2b) & MgCO_3 & + CaO & = MgO & + CaCO_3. & \\
(3) & CaSO_4 & + Na_2CO_3 & = CaCO_3 & + Na_2SO_4. & \\
(4a) & MgCl_2 & + CaO & = MgO & + CaCl_2. & \\
(4b) & CaCl_2 & + Na_2CO_3 & = CaCO_3 & + 2NaCl. &
\end{array}
$$

The bicarbonates constituting the temporary hardness are precipitated by the lime in the form of carbonate in the case of the lime hardness and as oxide from the magnesia portion (equations 1, *2a*, and *2b*). The sulphate of lime portion of the permanent hardness is decomposed by soda carbonate, its lime being precipitated as carbonate (equation 3), while the magnesium portion requires the conjoint action of the lime and soda carbonate to produce maximum elimination (equations *4a* and *4b*). To determine the exact quantities of the two reagents required for a particular water the following tests may be applied:

Lime.—210 c. c. of sample is placed in a stoppered cylinder and lime-water of known strength added in considerable excess, the mixture shaken during 2 hr., settled or filtered, 70 c. c. pipetted off and titrated with $\frac{N}{10}$ HCl first to phenolphthalein and then to methyl orange. The difference between the two is deducted from the figure obtained in the phenolphthalein titration. The result gives the lime present in excess of what is actually needed.

Soda.—70 c. c. of sample is placed in a platinum basin and excess of $\frac{N}{10}Na_2CO_3$ added, the solution is evaporated nearly to dryness and then somewhat diluted, and the precipitate washed with air-free water. The excess of soda carbonate in the filtrate is then found by titration with $\frac{N}{10}H_2SO_4$ (methyl orange).

The addition of the reagents to the water and separation of the precipitated sludge is carried out in a variety of appliances designed for the purpose by firms specializing in this line of engineering, and these work more or less automatically. It ought to be borne in mind, however, that natural waters are not constant in composition, and that for this and for other reasons such appliances will not continue to work satisfactorily without supervision. Much of the disappointment that has followed the installing of water-softening apparatus results from unreasonable expectations of what it can perform. Periodic testing of the treated water ought not to be omitted.

Boiler Fluids.—Where, for one reason or another, it is not thought

advisable to install a water-softening plant, the practice is sometimes followed of making additions to the contents of the boiler for the purpose of causing the scale-forming substances to separate as a soft sludge capable of being periodically expelled by blowing down. These so-called boiler fluids or anti-incrustation mixtures are of various compositions, sometimes consisting of a weak aqueous solution of soda carbonate or caustic soda alone, but more frequently containing, in addition, organic matter of a mucilaginous character (Irish moss, agar, starch, or gum), tannins, and other vegetable products belonging to the group of colloids. For small boiler installations and in special circumstances such compositions have their uses, but obviously they should be employed with caution, for reasons alike of safety and of economy.

Corrosion of Metals.—The destructive effect of certain waters on steam boilers has already been referred to and illustrates a subject of enormous importance to the engineer, namely the liability of metals in general to enter into chemical changes while in use, giving rise to what is called corrosion. In the rusting of structures exposed to moist air, the pitting of boilers, the giving out of steam-pipes, the erosion of turbine blades, and the failure of condenser tubes, it meets him on every side. In some cases the damage may be accidental, such as happens when an acid mine-water or a pickling effluent is admitted to a river or canal and is used for steam-raising, but for the most part it is the result of having to bring metals into contact with a system containing water, oxygen, and carbonic acid, reinforced in some cases by chlorides in solution, and these must be looked upon as more or less normal constituents of natural waters. The choice of metals for construction must be made with a number of considerations in view, of which chemical inertness is only one, and it frequently happens that mechanical qualities and cost decide the use in certain circumstances of a metal the composition of which renders it liable to considerable corrosion, in preference to one that is more stable.

Erratic Appearance of Corrosion.—One of the most puzzling features of corrosion as ordinarily observed is its erratic character, giving rise to a widespread belief either that it is a matter of chance or is to be ascribed to concealed defects in the metal for which the manufacturer is accountable. It is true that such defects sometimes exist, in the form of dissolved impurities, slag inclusions, imperfect mixing, and unsuitable crystalline structure, and that these may originate or accelerate wastage of the metal when in use; but it must be borne in mind that the purest commercial metals are not immune, and that care must be exercised in the selection of steels, brasses, bronzes, and other metallic mixtures to obtain what is best fitted to withstand the particular conditions they are to be subject to. As a matter of fact the erratic appearance of corrosion is due less to variations in the quality of the particular metal than to the number and variability of the chemical forces brought to bear upon it when in use. It may happen that a temporary circumstance of merely chance occurrence determines for a piece of metal whether its career is to be long or short, and the relative preponderance of protective and destructive forces

at one stage, by opening up or shielding the surface, may settle its behaviour at a later date.

Corrosion of Condensers.—The case of steam condensers has been the subject of prolonged study by a committee of the Institute of Metals and affords an illustration of the difficulty of dealing with corrosion in practice. The giving out of condenser tubes is nearly always due to the metal being attacked by substances in solution in the circulation water, and these are usually the same as cause corrosion in steam boilers. In the latter, if the trouble is serious, it can be dealt with by chemical treatment of the feed-water, but in the case of circulation water the enormous volume renders such treatment impracticable. In the case of steamers the difficulties are increased by the fact that sea- or river-water, often of doubtful quality, must be employed. These circumstances determine that instead of mending the water, amelioration must be sought in the direction of choosing a metal that will offer maximum resistance to attack, and of arranging details of construction and of operation so as to ensure the fewest opportunities of intensified action on the surface.

Choice of Metal for Condenser Tubes.—The prevailing practice is to manufacture the tubes of brass of the composition copper 70 per cent, zinc 30 per cent, but the purity of this mixture may vary according as electro-deposited or less refined metals have been employed. Where corrosion is troublesome or the conditions are likely to be severe, the substitution of Admiralty alloy of composition copper 70 per cent, zinc 29 per cent, and tin 1 per cent is suggested. In either case the iron should if possible not be allowed to exceed 0·1 per cent. A brass containing 2 per cent of lead has been found serviceable, and in the case of acid waters the employment of an alloy of 80 per cent copper and 20 per cent nickel or of an arsenical copper may be necessary. The crystalline structure of the drawn metal is susceptible of improvement as regards resistance to corrosion, and this can be obtained to some extent by subjecting the tubes to an oxidizing annealing for 3 hr. at ($660 \pm 45°$ F.). The annealed tubes should not be subsequently pickled.

Effect of Lodgments.—The combined action of water, air, chlorides, and carbonic acid upon the brasses gives rise to mixed basic salts of copper and zinc of variable composition and colour. Sometimes these appear to act as a protective covering; others, such as cuprous chloride, behave differently and actively promote oxidation. If at any point the latter become attached to the surface, a centre of corrosion is established and pitting will probably ensue. Any lodgment of foreign matter, even though itself chemically inert, obviously favours the attachment of such activating substances and may thereby promote local corrosion. Increasing the speed of the water current suggests itself as a means of preventing or lessening such deposits, but on the other hand the risk is thereby incurred of dislodging protective scale which may have formed on the surface and be a very efficient safeguard, so that the remedy must be used with judgment.

Foaming Waters.—Conditions that give rise to foam intensify cor-

rosion. When the water is churned so that entrapped air bubbles are emulsified, a very active chemical agent is thereby created and the action of the dissolved oxygen is powerfully reinforced. Throttling of the water current at sharp bends or at constrictions may establish such conditions and should be guarded against. The stability of foams is increased by the presence of bodies that lower the vapour tension, and such bodies are found in the oily and certain other contaminations of estuary waters and in the gelatinoid substances in sea-water. Increased temperature of the circulation water, such as prevails in tropical areas, and local augmentations, as may be occasioned by the introduction of auxiliary steam into the condenser, add to the risk of corrosion. On the other hand even moderately hard waters give rise to coatings of scale, largely sulphate of lime, which, provided they are slowly formed, exercise a markedly protective influence on the surface. New tubes, from which such scales are absent, are specially prone to attack, and corrosion that is then initiated may be difficult to arrest even when the original agent has been removed. Special care is therefore called for in the safeguarding of new tubes until a protective skin has been acquired.

Methods of Protection.—On the supposition that corrosion is essentially an electro-chemical action, methods have been proposed and are in use for the protection of condenser tubes. These are based on the employment of counter-currents of electricity which may be produced either by steel blocks screwed into the tube plates and forming an electric couple with the metal to be protected, or by stronger currents specially generated from a dynamo or battery. Aluminium blocks have also been tried with success. Such methods, where carefully carried out, give a measure of protection, and in numerous cases have resulted in lengthening considerably the life of the tubes. They are not, however, to be looked upon as an infallible barrier to corrosive action. The number and variety—as well as the variability—of the contributory factors militate against any single solution being thoroughly effective at all times, and meantime a study of the individual circumstances seems an indispensable prerequisite to improvement in each case, to be followed by such modifications as that study may suggest.

LUBRICANTS

The object of lubrication is to minimize friction between moving surfaces and to prevent a consequent rise of temperature. When a journal is rotating rapidly on a well-lubricated bearing the two are separated by a continuous film of liquid. In such circumstances the particles of the oil keep rolling over one another, and resistance to this movement constitutes the viscosity of the oil. The friction of a liquid is due to its viscosity, and when used under these conditions, mineral and fixed oils of identical viscosity are equally efficient lubricants. On the other hand, when a bearing is subject to a heavy thrust pressure, the oil film on the face is not continuous and friction is consequently high. In this case another property of oils than viscosity plays a part, namely the adhesion of the particles of the oil to the

metallic surface, whereby it is able to spread itself over the latter and to resist removal and dispersion. This quality is designated " oiliness " and may be defined as that property of a lubricant in virtue of which it maintains an unbroken film under a heavy load. In respect of oiliness mineral oils, which are hydrocarbons, are inferior to fixed oils, which are compounds of an alcohol and an acid, and for this reason mineral oils are improved as lubricants for ordinary machinery by blending with fatty oils.

Classification of Lubricants.—Lubricants may be divided into the following groups:

1. Vegetable and animal oils and fats and liquid waxes. Examples of this group are rape, castor, and sperm oils.
2. Mineral oils, derived from the fractions of petroleum and shale oil boiling above 570° F.
3. Blended oils, prepared from mineral oils with an admixture of vegetable or animal oil. Boiled vegetable oils are sometimes incorporated with mineral oils to give these the consistency necessary for heavy machinery.
4. Solid lubricants, including graphite, French chalk, or mica, with or without addition of grease. Greases used alone may consist of natural fat or of a mixture of mineral or fatty oil with a soap compounded of lime or aluminium with a fatty or resin acid. Aquadag and oildag are preparations of artificial graphite in the colloidal state dispersed in water or in oil.
5. Lubricants for use with cutting tools, and consisting usually of a solution of soft soap and soda in water.

Testing of Lubricating Oils.—The following physical tests are commonly relied upon in determining the quality of a lubricating oil and its suitability for a particular purpose:

1. Specific gravity.
2. Viscosity.
3. Flash-point.
4. Setting-point.

The property of oiliness is meantime not readily subject to direct measurement and can be tested only comparatively on a suitable machine. For the determination of *specific gravity* the specific-gravity bottle is in general use. The Westphal balance and the Sprengel tube are also employed. *Viscosity* is determined in practice by the time required for a definite volume at a given temperature to pass through a standard opening under a specified head or pressure. Redwood's viscometer is the recognized instrument in use for the purpose in this country, and the viscosity at a given temperature is the number of seconds required for 50 c. c. to emerge from the orifice of a standard cup filled to a definite height. Determinations are done at 60° F. (if possible), and at two higher temperatures suggested by the conditions under which the oil is to be used.

The *flash-point* is the temperature at which an oil gives off vapour fast enough to produce an explosive mixture with air either in an open cup (open

test) or in a confined space (close test). For lubricating oils the close test may be made either by Gray's or the Pensky-Martin instrument, in which the oil is heated in a closed cup which is momentarily opened to allow contact with a flame. A thermometer dips into the oil, and uniformity of temperature is maintained by constant stirring with a paddle. A flash-point not under 350° F. is desirable in lubricating oils for use inside buildings.

The *setting-point* is the temperature at which an oil becomes stiff and ceases to flow. The sample in a corked test-tube is cooled either in water or in a freezing mixture until it remains in position on reversing the tube. The temperature should be held constant for about twenty minutes before observing the oil.

Chemical Tests.—For more complete characterization of a lubricating oil a chemical examination is necessary to supplement the physical tests. This generally includes some or all of the following:

1. Examination for mineral acid.
2. Determination of suspended matter.
3. Gumming and volatility tests.
4. Determination of total acidity.
5. Determination of unsaponifiable matter.
6. Iodine value.

To test for mineral acid, which if present would actively corrode metal bearings, a weighed quantity is vigorously agitated with water, allowed to stand, and a drop of methyl orange solution added. A pink colour indicates mineral acid which may be estimated by titration with alkali.

Suspended matter is separated by thinning a weighed sample with ether and passing through a weighed filter which is afterwards carefully washed with ether till free of oil and dried at 212° to 220° F.

Volatility is a property which in the case of mineral-oil lubricants must be carefully watched as, in contrast to fatty oils, these may contain a proportion vaporizing at moderate temperatures. Archbutt's test consists in subjecting 0·5 gm. of the oil placed in a shallow boat within a heated tube to a current of air passed at the rate of 2 litres per minute for one hour. The air is previously heated to the desired temperature.

Gumming may be tested for by warming 1 gm. on a watch-glass for twelve hours in a boiling-water oven. Liability to decomposition is a property that under certain conditions must be taken into consideration. For cylinder lubrication in high-pressure steam and in gas- and oil-engines pure fatty oils are unsuitable. For the former a blended oil or a mineral oil of suitable viscosity may be used; for the latter pure mineral oils only are to be preferred of a type not readily carbonized on heating.

The more elaborate chemical tests for lubricants are those in use for examination of oils in general. For details of working reference may be made to the special textbooks dealing with the analysis of oils and fats.

THE RECIPROCATING STEAM-ENGINE

BY

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The Reciprocating Steam-engine

Introduction

The steam-engine is a prime mover, which may be defined as any contrivance for the production of mechanical work by utilizing one of the non-vital forces of nature. The force must be exerted against a yielding resistance, and, in general, it may be regarded as a pressure. The product of the pressure, and the distance through which the resistance is moved at the point of application, is a measure of the work done.

The ancients understood that fluid pressure could readily be obtained by the application of heat to a closed vessel containing water, the vapour produced exerting a pressure in all directions on the sides of the vessel, and that the vapour might be condensed by the application of cold, leaving a vacuous space above the cooled water.

A number of experimenters endeavoured to make use of these physical facts, but the first to produce a working apparatus was Savery, who, towards the end of the seventeenth century, patented an arrangement for pumping water.

Briefly described, it consisted of a boiler, in which high-pressure steam was generated, and a vessel communicating, at the lower portion, with a suction pipe and a rising discharge main. Steam from the boiler was admitted to the vessel, driving out the air through the discharge valve. The steam was shut off, and then condensed by allowing cold water to flow over the outer surface of the vessel. As soon as the pressure inside became sufficiently reduced, water from the suction pipe began to flow in, completing the condensation, and when the vessel was judged to be full of water, steam was again admitted, and the water forced out through the discharge valve. The steam contained in the vessel was then condensed as before, and the process repeated. The steam supply was regulated by hand, the suction and the discharge valves being automatic in action, exactly as in an ordinary plunger pump.

This simple apparatus, which could hardly be called an engine, was exceedingly wasteful. The steam was in direct contact with the water to be pumped and with the sides of the vessel containing the cold water, and much of it was condensed uselessly. The height to which the water could

be forced depended upon the pressure of the steam, and it is interesting to note that Savery occasionally employed pressures as high as eight to ten atmospheres, in copper vessels with joints of hard solder, and, at first, without a safety valve.

The next step in the development of the steam-engine was made by Newcomen, who employed a vertical cylinder with a piston to drive pumps fixed in the mine shaft.

A wooden beam was fixed above, the ends of which formed sectors of a circle struck from the centre of the gudgeon upon which the beam oscillated, the piston-rod at one end and the pump-rods at the other being connected to the beam by flat chains, which were alternately wound and unwound on and off the sectors during working. The cylinder was open at the top, the upper surface of the piston being in contact with the atmosphere. Assuming the piston to be at the bottom of its stroke, steam at about atmospheric pressure was admitted below. The piston rose and the pump-rods descended, allowing the barrel to fill with water. Communication with the boiler was then cut off, and a jet of water from an elevated tank was allowed to enter the cylinder, thus condensing the steam and forming a vacuum. The piston was then forced downwards by the pressure of the atmosphere and the pump-rods raised, lifting the water out of the pump barrel to the surface. Steam was then admitted to the cylinder, allowing the piston to rise, and the products of condensation were forced out of the cylinder through a "snifting" valve.

The efficiency of the Newcomen engine was not greatly superior to that of Savery's. Exactly the same fundamental faults existed. The steam was not used expansively, but was employed merely as a means for producing a vacuum, and was allowed to enter a cylinder which had just previously been cooled by the presence of water. Nevertheless, it was a great step in advance. Newcomen had produced a machine, not merely an apparatus. The depth from which water could be raised depended not upon the pressure of the steam, for he used very low pressures, but only upon the diameter of the piston and the length of stroke. Later the engine was improved greatly in detail, especially by Smeaton. All the necessary movements of the various valves were made automatic, and it became thoroughly workmanlike and reliable, so that it was extensively adopted for the pumping of water; but just about this period the improvements effected by James Watt caused it to be rapidly superseded, although survivals of the type were in use until comparatively late in the nineteenth century.

In 1763, James Watt, whilst engaged in repairing a model Newcomen engine used for demonstration purposes at Glasgow University, perceived the great waste caused by the condensing portion of the cycle being allowed to take place in the working cylinder, and he conceived the idea of using a separate vessel as a condenser. He also saw the necessity for removing the air carried in by the condensing water and by leakage, which, of course, impaired the vacuum by its expansion from atmospheric pressure. A pump for removing both the air and products of condensation was employed by

him for the first time, and has remained a permanent and indispensable feature of steam prime movers of all types when working condensing.

Further improvements, with the object of conserving the heat in the steam, were made by Watt, who was the first to comprehend that the steam cylinder should be kept as hot as the steam which enters it. The top of the cylinder was closed by a cover through which the piston-rod passed, a stuffing box being used. A steam jacket was added, and the whole cylinder was lagged with non-conducting material, in order to prevent loss of heat by radiation. Steam was allowed access to the whole space above the piston, with the object of keeping the cylinder warm, its pressure performing the same function as the atmosphere in Newcomen's engine. Watt's first engine, therefore, worked on a similar cycle, the sole difference being that the working steam from the under side of the piston escaped from the cylinder at the termination of the upstroke and was condensed in a separate vessel.

Later, expansive working of the steam was introduced, again raising the efficiency, and the engine was made double-acting, that is, each side of the piston was put into communication alternately with the boiler and the condenser. These changes practically completed the series of wonderful improvements introduced by the genius of Watt in the use of steam as a working fluid in a heat engine, and it is interesting to note that they were effected in complete ignorance of the equivalence of heat and work, which long afterwards was established by Joule.

Other inventions, not less valuable, followed, such as the parallel motion, the centrifugal governor, the indicator, &c., all of which aided the development of the steam-engine as a working machine. The slide valve was introduced by Murdoch, one of Watt's assistants. The application of the crank and connecting-rod brought the engine into general use for all purposes for which rotative motion is necessary, and led to a world-wide development in industries of all kinds.

A new cycle was introduced in the large engines used in Cornwall for the pumping of water from the mines, and therefore was called the "Cornish" cycle. The working stroke was downwards, steam being admitted from the boiler above the piston and cut off comparatively early. At the bottom of the stroke the steam above the piston was allowed to pass to the under side by a so-called equilibrium valve being opened, so that by the time the upstroke was completed the whole of the steam had been transferred from the top side to the under side of the piston. At the commencement of the next downstroke a valve was opened, which allowed the whole of the steam below the piston to flow into the condenser, thus causing a vacuum to be formed below the piston. By this artifice the upper portion of the cylinder was never in communication with the condenser, and the process of "initial condensation" was thereby reduced, giving much improvement in economy, so that the Cornish pumping-engine attained an efficiency rivalled only by high-pressure multiple-expansion engines of to-day, if the systematic records of the performance

of numerous engines of that type are to be believed. The cycle, of course, was of the " single-acting " type, and it is interesting to note that the celebrated Willans engine operated upon a similar cycle, with the difference that the expansion of the steam took place generally in more than one cylinder.

All the types of engines just described worked with steam of very low pressure, and it is remarkable that although Watt was well aware of the economy given by expansive working, yet he steadily resisted the tendency to adopt higher boiler pressures, many of his engines working at a pressure of about 7 lb. per square inch. Due principally to the work of Trevithick, pressures gradually rose, and the high-pressure non-condensing engine was introduced.

The next notable improvement was made by Hornblower, who invented the compound engine.

From a thermal point of view the chief advantage given by compounding is that the total range of temperature from the stop valve to the exhaust is divided into two or more stages, so that the variation in any given cylinder is less, and the losses caused by initial condensation are reduced. In modern engines, compounding is carried through two, three, or, in the case of some marine engines, even four cylinders, the initial working pressures with the latter being as high as 220 lb. per square inch, and at the present moment it is proposed to increase the boiler pressure to 300 lb. per square inch, and to divide the expansion of the steam into five stages.

The latest development in the thermal design of reciprocating steam-engine is known as the " Uniflow ". An entirely new cycle is used, which enables the steam to be expanded in one cylinder only, with the same economy as is obtained with engines of the triple-expansion type.

With the usual arrangement the steam enters and leaves by the same passages, or, in the cases where there are separate inlet and exhaust ports, at the same part of the cylinder. The piston, cylinder covers, and the surfaces of the ports are in contact with, and are scoured by, the exhaust steam during the whole of the return stroke. They are cooled much below the temperature of the inlet steam, with the consequence that a portion of this incoming steam is condensed, and therefore the following stroke commences with a mixture of steam and water in the cylinder.

As the pressure and temperature of the steam fall during expansion, some of the water is re-evaporated, partly by its own heat and partly by taking heat from the cylinder walls, but much of the heat restored escapes unused to the exhaust. In the case of a compound engine it would, of course, be used in the next cylinder, but from a thermodynamic point of view this process is very wasteful, as the heat is abstracted from the steam at the temperature of admission and restored at a lower temperature.

Fig. 1 shows an indicator diagram from a single-cylinder engine. Volume V shows the amount of steam present at cut-off as disclosed by the indicator. Volume V_1, however, shows the actual amount of steam which is present as a mixture of steam and water, therefore $\frac{V}{V_1}$ shows the

fraction of the contents which consists of vapour, and is called the " dryness fraction " at cut-off. The quantity $V_1 - V$ is called the " missing quantity ", because it is not shown by the indicator. The total quantity V_1 entering the cylinder during each admission period is arrived at by consumption tests taken by condensing the exhaust steam and weighing it. The missing quantity is explained partly by the condensation which takes place during admission, the metal of the cylinder and ports having been previously cooled by the exhaust steam, as explained above.*

The dotted curve is the saturation curve. To obtain points on the curve, take from steam tables the volumes corresponding to various pressures between cut-off and release. Multiply these volumes by (the cylinder feed per stroke + weight of cushion steam). This gives values for V_1 in saturation curve. The horizontal distance at any pressure between corresponding points on the expansion curve of the actual indicator diagram and the saturation curve shows the amount of water present, the quantity of steam in the cylinder being assumed to be constant between cut-off and exhaust. In a real engine this condition may not exist, as there may be leakage of steam either inwards or outwards through the valves or past the piston.

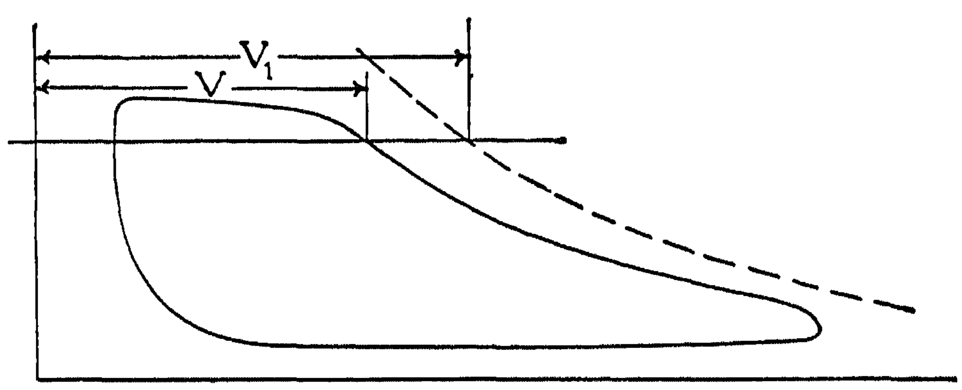


Fig. 1.—Single-cylinder Engine Indicator Diagram

A means of reducing the initial condensation, which has been revived and widely used latterly, consists in heating the steam to a temperature higher than that of saturation. Superheated steam will not condense until cooled through a range of temperature equal to the amount of superheat. The colder cylinder metal has less effect, so that the steam tends to remain dry during admission proportionately to the degree of superheat. The thermodynamic equivalent of the available part of the heat added to the steam by superheating would account for a fraction only of the improvement in steam consumption actually obtained. Valve leakage is also considerably reduced.

The Uniflow type of engine was first proposed by L. J. Todd in his patent of 1887, and his specification shows that he was well aware of the thermal defects of the ordinary type and of the modifications necessary for improvement. He stated that the object of his invention was to " produce and maintain an improved graduation of temperature extending from each of the two hot inlets to the common central cold outlet, which shall cause less condensation of the entering steam ".

The cycle is as follows. Steam enters through the admission valve, and is then cut off at a predetermined point of the stroke in the usual way. Expansion commences and is continued until about 90 per cent of the stroke

* Other factors affecting the missing quantity are piston and valve-leakage (see Callendar and Nicholson, *Proc. Inst. C. Eng.*, Vol. CXXXI, 1897).

has been completed, when the piston uncovers a ring of ports in the wall of the cylinder which communicate with the condenser, the pressure in the cylinder thus falling to very nearly that in the condenser. On the return stroke the ports are closed by the advancing piston, and the steam left in the cylinder is compressed until it has attained a pressure nearly equal to that of the incoming steam. The process of compression raises the temperature of the steam, the cylinder walls, the cylinder-cover faces, and the piston. As the steam enters at one end of the cylinder and leaves at what may be called the other end, there is a gradual fall of temperature in the cylinder metal from the inlet end. Further, the steam when exhausting does not scour the surfaces at the inlet end and so reduce their temperature by abstracting heat, although the temperature must fall to some extent by adiabatic expansion during the brief exhaust period, but heat is restored by the subsequent compression. Some of the heat generated by compression must be lost by conduction, and the work expended is not wholly recovered in the subsequent power stroke; but the total effect of the arrangement is that initial condensation is greatly reduced and the efficiency improved, compared with single expansion in the ordinary type of engine.

It has been seen that there are many disturbing factors which interfere with the ideal performance of a steam-engine, such as initial condensation, re-evaporation, leakage, conduction, and radiation, all leading to uncertainty with regard to the conditions under which the expansion of steam takes place. This uncertainty, taken together with the fact that the cycle is neither complete nor reversible from a thermodynamic point of view, makes a rational theory difficult, and perhaps unattainable.

The injunction to keep the steam warm, and the avoiding of throttling and large clearances, embody the whole philosophy and practice of design from the thermal point of view, and little progress has been made since Watt laid down the axiom "that the cylinder should be kept as hot as the steam which enters it". Having borne these considerations in mind, designers can do little more, and have been perforce contented with the possibility of ascertaining and checking the performance of their engines by the method of weighing the condensed steam, introduced by Willans. The ease with which it is possible to measure electrical loads has made universal the practice of submitting to this test both steam-engines and turbines which drive electrical generators, so that makers are now able to guarantee steam consumptions within very narrow limits.

Current practice is to refer the performance of a steam-engine to that which would be given by an ideal engine working through a special cycle proposed by Rankine. The steam is supposed to be admitted at full steam-chest pressure to the point of cut-off and then expanded adiabatically, that is, without loss or gain of heat, to the back pressure, the steam remaining at that pressure throughout the exhaust stroke. The engine is not supposed to have any clearance between the piston and cover, or in the ports, and there are no losses of any kind. Under those conditions 1 lb. of steam is

capable of performing a definite and calculable amount of work, which consists of the work done before cut-off plus the exact equivalent of the difference between the heat contained in the steam at cut-off and the heat which it contains at exhaust.

Fig. 2 shows an ideal Rankine cycle (EFGH) around the actual cycle ABCD. In an actual engine, a diagram such as indicated by the inner line would be obtained. In the first place there would be loss of initial pressure during admission, due to the throttling in the ports and past the valve edges, as shown by the sloping line. Some of the steam would be condensed as explained before, and the volume of steam at cut-off would be less than that shown by the Rankine diagram. The steam would then expand

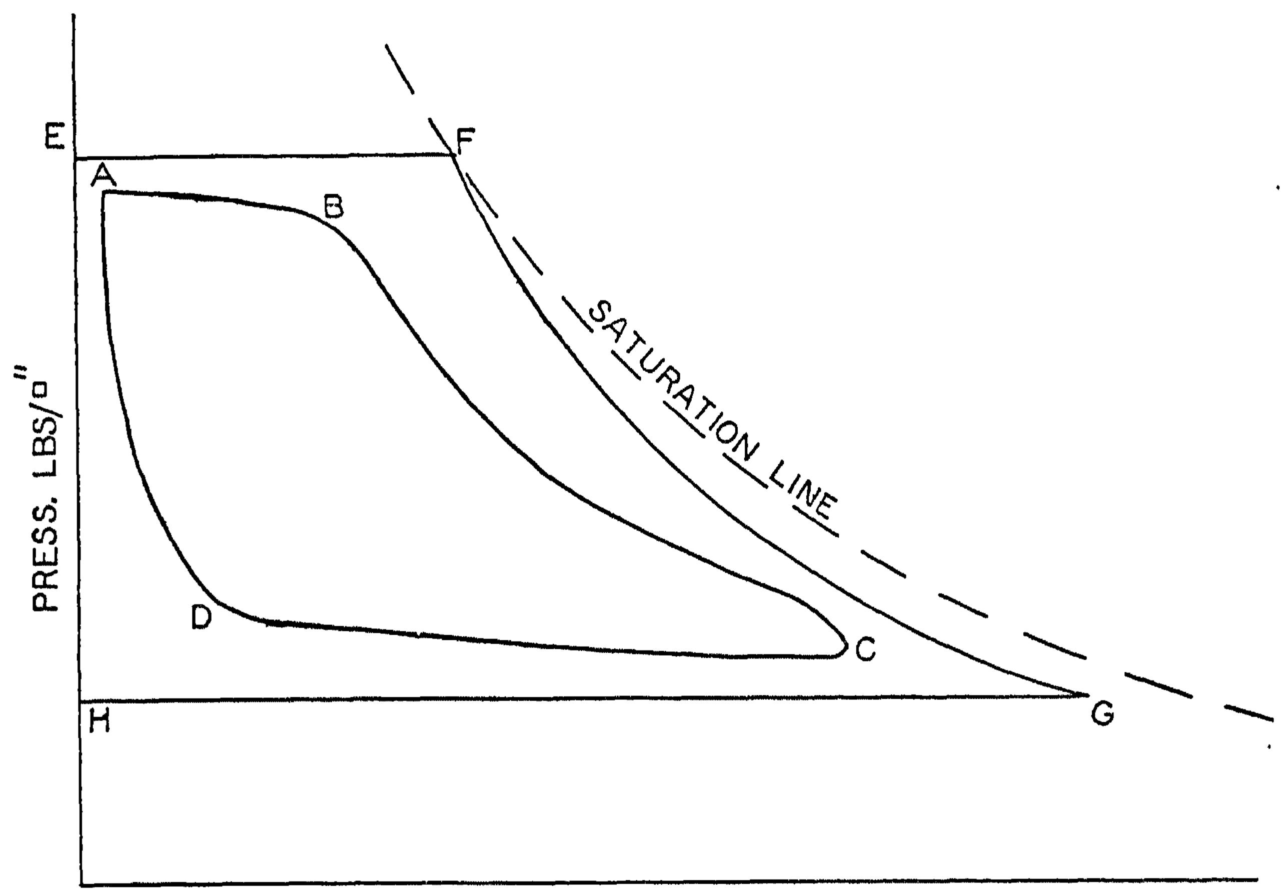


Fig. 2.—Indicator Diagram with Rankine Cycle

under the disturbing influence of the presence of water and the effects of conduction and radiation, and expansion would not be continued to the line of back pressure, especially in the case of condensing engines, owing to the impracticably large cylinder necessary, which would cause more work to be absorbed by friction than would be given by complete expansion. The pressure of the steam in the cylinder during the exhaust stroke would be higher than the back pressure due to the effect of friction through the ports, and there would be some compression at the end of the stroke into the clearance volume.. This volume is represented by the horizontal distance between the beginning of the admission line and the line of zero volume in the Rankine diagram.

The difference in area between the two diagrams shows the losses which take place in a real engine, and is a measure of its inefficiency compared with an engine working on a Rankine cycle.

A saturation curve shown by broken lines has been added to fig. 2. The horizontal distance at any point between this curve and the expansion curve of the Rankine diagram indicates the progressive wetness of the steam due

to adiabatic expansion in the ideal engine with a perfectly non-conducting cylinder.

The instrument called the indicator, by which a graphical record of the cyclic variation of pressure in a steam cylinder is made, was invented by Watt. He used a board upon which a sheet of paper was fixed, the board being given a to-and-fro movement less in extent but exactly proportional at all points to the movement of the piston. A small steam cylinder containing a piston loaded with a spring, the rate of compression of which was accurately known, was fixed in such a way that the movement of the piston was at right angles to the direction of movement of the board. A pencil was fixed to the piston-rod, and on steam being admitted from the main cylinder to the under side of the indicator piston the latter moved in such a way as to cause the pencil when applied to the paper to trace a curve which at any instant showed the pressure of the steam in the engine cylinder. The height of the diagram traced depended of course upon the relation of the force exerted by the spring to the area of the indicator piston.

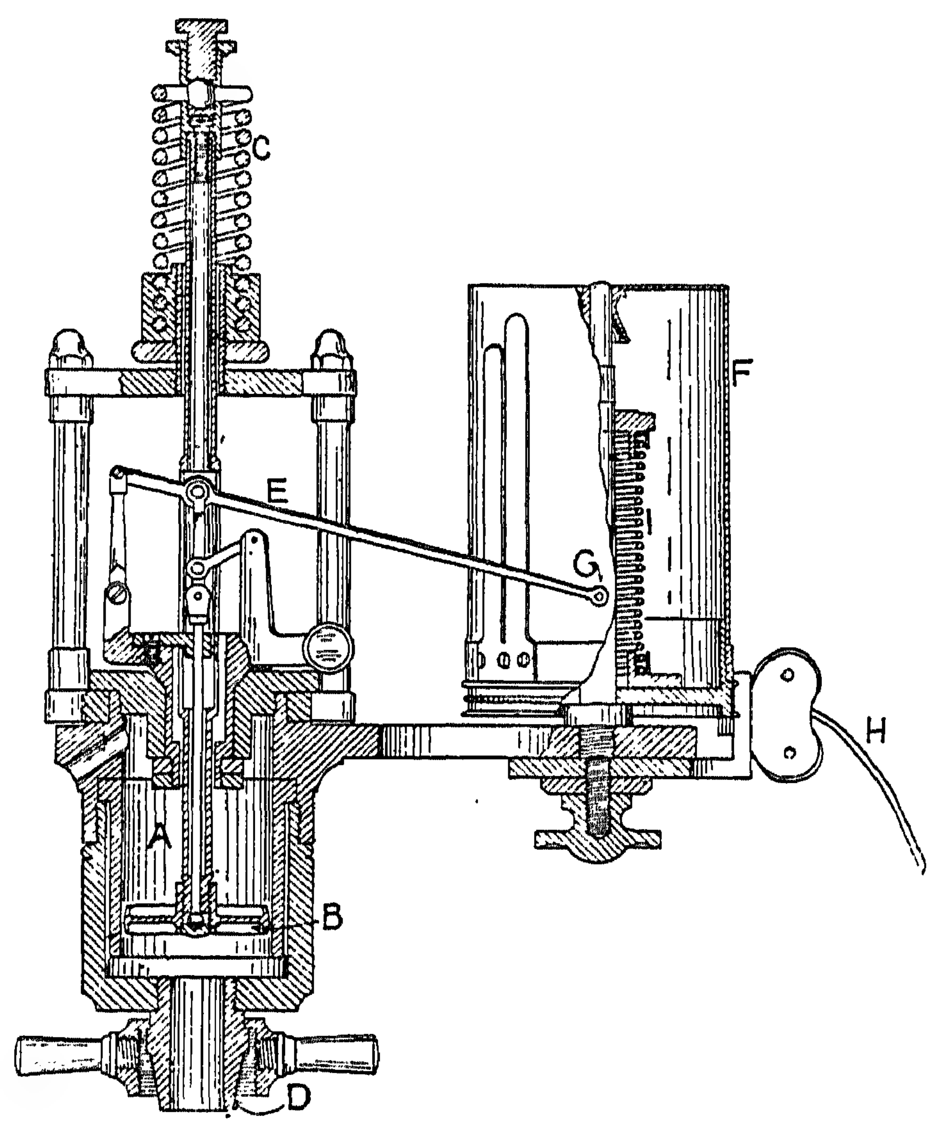


Fig. 3.—Crosby Indicator

The area of the diagram thus represents the quantity of work performed during one stroke of the engine, the method usually adopted being to find the mean pressure from the diagram and multiply this by the area of the engine piston in square inches and by the length of the stroke in feet, the product giving the number of foot-pounds of work developed during the stroke.

In the modern indicator the sliding board is replaced by a drum, to which is given a rotary to-and-fro movement by a cord attached at one end to a part which repeats the motion of the crosshead on a reduced scale, the other end of the cord being alternately wound and unwound on and from the drum, the motion of which is controlled by a clock-spring. The paper is wrapped round the drum and kept in position by two spring clips. A metallic pencil is used, for which the surface of the paper is specially prepared.

Fig. 3 shows the instrument manufactured by Messrs. Crosby & Co., Ltd. A is a small cylinder; B an accurately fitted piston, having an area of usually 1 sq. in.; C the spring; D a joint, by means of which the instrument can be attached to the cock on the engine cylinder; E a straight-line mechanism, by which the movement of the piston B is repeated on an

enlarged scale at the pencil G which consists of a short length of soft brass wire. The paper is carried on the drum end to which a rotational to-and-fro motion is given by the cord H, which is kept taut by the resistance of the spring I contained in the drum F.

On steam being admitted beneath the piston, the pencil G rises and falls as the pressure of the steam varies, thus tracing a curve on the paper. The curve is closed, as the changes are cyclic and steady.

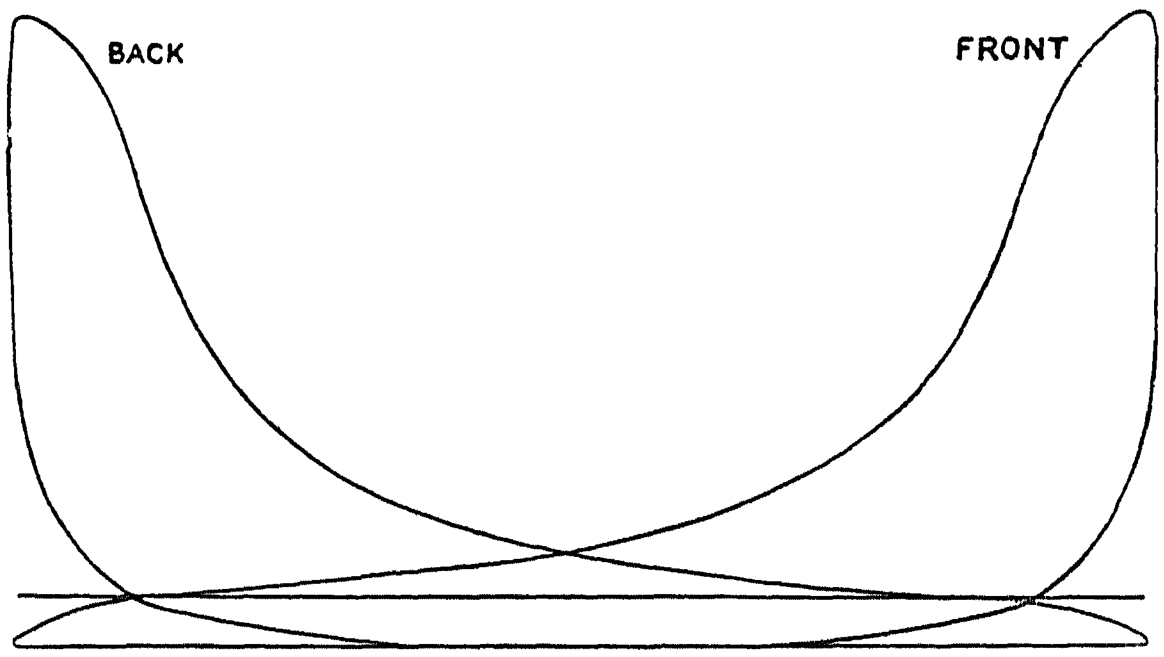


Fig. 4.—Indicator Diagram taken from Uniflow Engine

Fig. 4 shows a diagram taken from a Uniflow engine made by Messrs. Robey & Co., Ltd.

To ascertain the mean effective pressure of the steam in the cylinder, an instrument called a planimeter is used, by which the area of the diagram is obtained. This area, divided by the length of the diagram and multiplied by the scale of the spring, gives the mean effective pressure.

When a planimeter is not available, the diagram may be divided into ten parts, as shown by fig. 5, the two end divisions being half the width of the others.

Obviously the effective pressure on the piston is the difference between

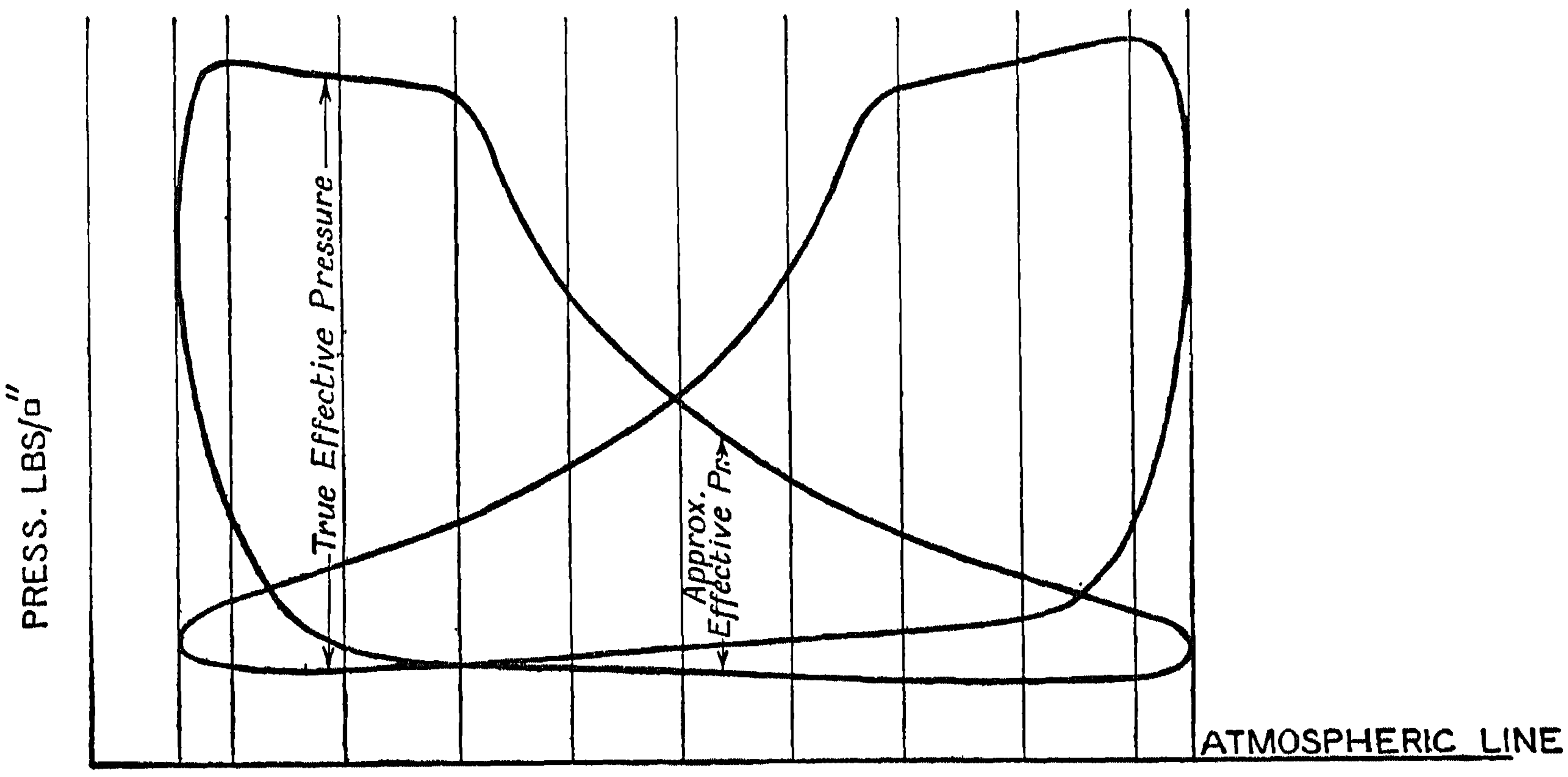


Fig. 5.—Graphic Method to obtain Mean Effective Pressure of Indicator Diagram

the absolute pressures on the steam side and the exhaust side at any instant, and is given by measuring the depth at any point between the steam line of one diagram and the exhaust line of the other. An approximate method is to take the difference between the steam and exhaust lines of *each* diagram instead of the steam line of one and the exhaust line of the other (see fig. 5).

Also, when a planimeter is used, the area of each diagram is obtained

separately, by following the outline with the tracing point of the planimeter.

Whichever of the methods of graphical measurement is adopted, the total length of the division lines divided by ten and multiplied by the scale of the indicator spring will give the mean effective pressure for each stroke. The average of the two mean pressures must be taken, and the indicated horse-power can be computed as follows:

Let A equal area of cylinder in square inches, P the average mean pressure in pounds per square inch for both strokes, L the length of stroke in feet, N the revolutions per minute, then I.H.P. $= \dfrac{2PLAN}{33{,}000}$.

CHAPTER I

Stationary Engines

Introduction.—Stationary engines are used mainly for the generating of power for industrial purposes, and several examples of large units will be described and illustrated, but more attention will be given to the latest development known generally as the Uniflow type, which has achieved such signal success that several firms who have long been identified conspicuously with the manufacture of what were the prevailing types, have almost entirely abandoned them in order exclusively to take up that of the Uniflow type.

Fig. 6 shows the usual arrangement of cylinder, &c. The working barrel is a separate casting furnished with deep cellular covers in which the admission valves are placed. The steam inlets are in the covers, which are thus jacketed by high-pressure steam. The valves and seats are fixed in the covers quite close to the internal surface, so that the ports are short and the clearance volume is small. There is a ring of exhaust ports at the centre of the length of the barrel, which communicate with the exhaust belt. The collective area of these ports may be made large, and if desirable the condenser, when of the jet type, may be placed immediately below the cylinder exhaust branch, so that the difference in pressure between the condenser and the cylinder at exhaust could be very small.

With a surface condenser it is usual to interpose an oil separator, in order to prevent the tubes becoming coated with a film of oil, and thereby prevent oil getting into the boilers when the condensate is used for boiler feed. Sometimes a feed heater is placed immediately below the exhaust branch. The axial length of the exhaust ports is usually about 10 per cent of the stroke, so that the piston, which itself forms the exhaust valve, has a length of 90 per cent of the stroke. Compression takes place, of course, during 90 per cent also of the stroke.

The total volumetric clearance is very low, varying from $1\frac{1}{2}$ to 2 per cent according to the size of engine. Assuming a clearance volume of 2 per cent, the ratio of volumetric compression would then be 46, so that the pressure of the entrapped steam may be made almost equal to that of the inlet steam, the temperature of the steam and of the metal surfaces being raised at the same time. This feature also tends to reduce initial condensation. There must, of course, be some loss, as the compressed steam will lose heat by conduction in this process, and cannot therefore give out on the return stroke the same amount of work as was spent upon it, but the whole effect seems to be in favour of economy.

So large a ratio of compression cannot be obtained in ordinary engines because of the large clearance volumes, and it is very necessary that the inlet valves be kept perfectly tight. Even a small leakage of high-pressure steam into the cylinder from the steam chest during the compression stroke would result in a high final pressure, with a diminution of the work area of the indicator card. An impaired vacuum would have a similar effect, and the blowing of the cylinder relief valves would call attention to the trouble.

For full load the cut-off is only about 8 to 10 per cent of the piston stroke, and for 25 per cent overload a cut-off of only about 14 per cent would be required, so that this type of engine readily responds to varying power demands.

When working non-condensing, when starting up, and again when stopping, it is necessary to relieve the pressure in the cylinder during the compression stroke, to prevent the see-sawing action which might take place, and cause the belts on lines of shafting driven by the engine to be thrown off the pulleys.

There are various devices for relieving the pressure. A special valve is provided which allows the compressed steam to enter a space in the cylinder cover, thus temporarily increasing the clearance volume. In other cases the steam in the cylinder is by-passed to the exhaust by an automatic arrangement worked by the pressure in the exhaust belt. The arrangement adopted by Messrs. Robey & Co. is shown in fig. 7. The valve in the cover is operated by a cam on a rocking shaft driven by an eccentric mounted on the valve-gear lay shaft, through a clutch which is put into gear by the spring-controlled piston working in a cylinder in communication with the exhaust. When there is a poor vacuum or when the engine is working non-condensing, the clutch is put into gear by the spring and the cams operate the relieving valves, but when condensing, the clutch is disengaged by the vacuum, the cam-shaft is stationary, and the valves remain closed. Messrs. Cole, Marchent, & Morley use their steam ejector air pump of the Delas type for starting up, by creating a vacuum in the condenser before admitting steam to the engine.

Where considerable quantities of heating steam at moderate pressures are required for process work, a compound engine is often employed, consisting of an ordinary high-pressure cylinder and a low-pressure Uniflow

cylinder, the steam for heating being taken off from a receiver between the two cylinders.

The steam consumption per indicated horse-power of the Uniflow

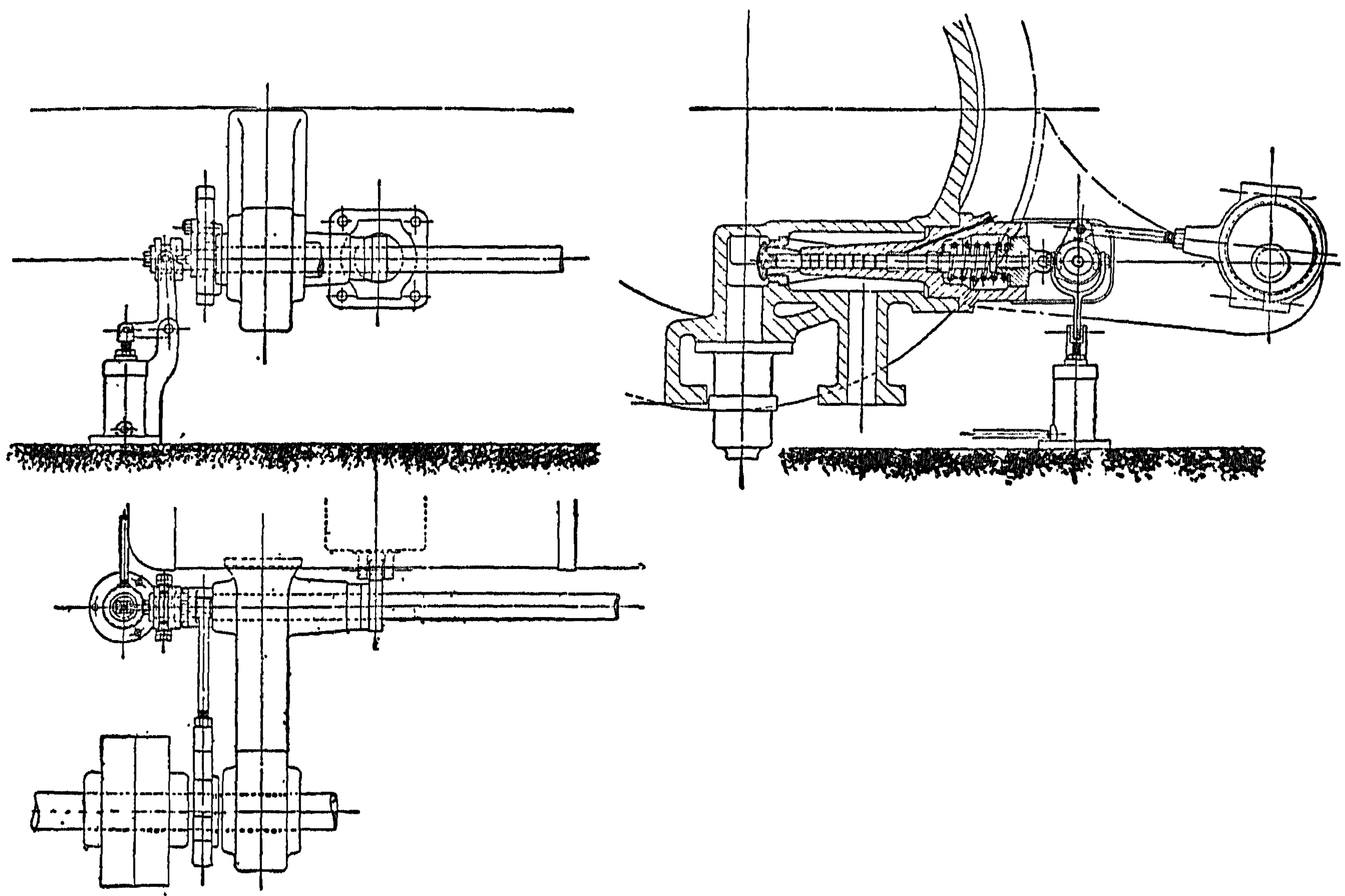


Fig. 7.—Robey Automatic Pressure-release Arrangement

engine is very low, and it is claimed that an economy equal to the best design of triples can be obtained under similar conditions. Fig. 8 shows a curve taken from a test of a Sulzer engine.

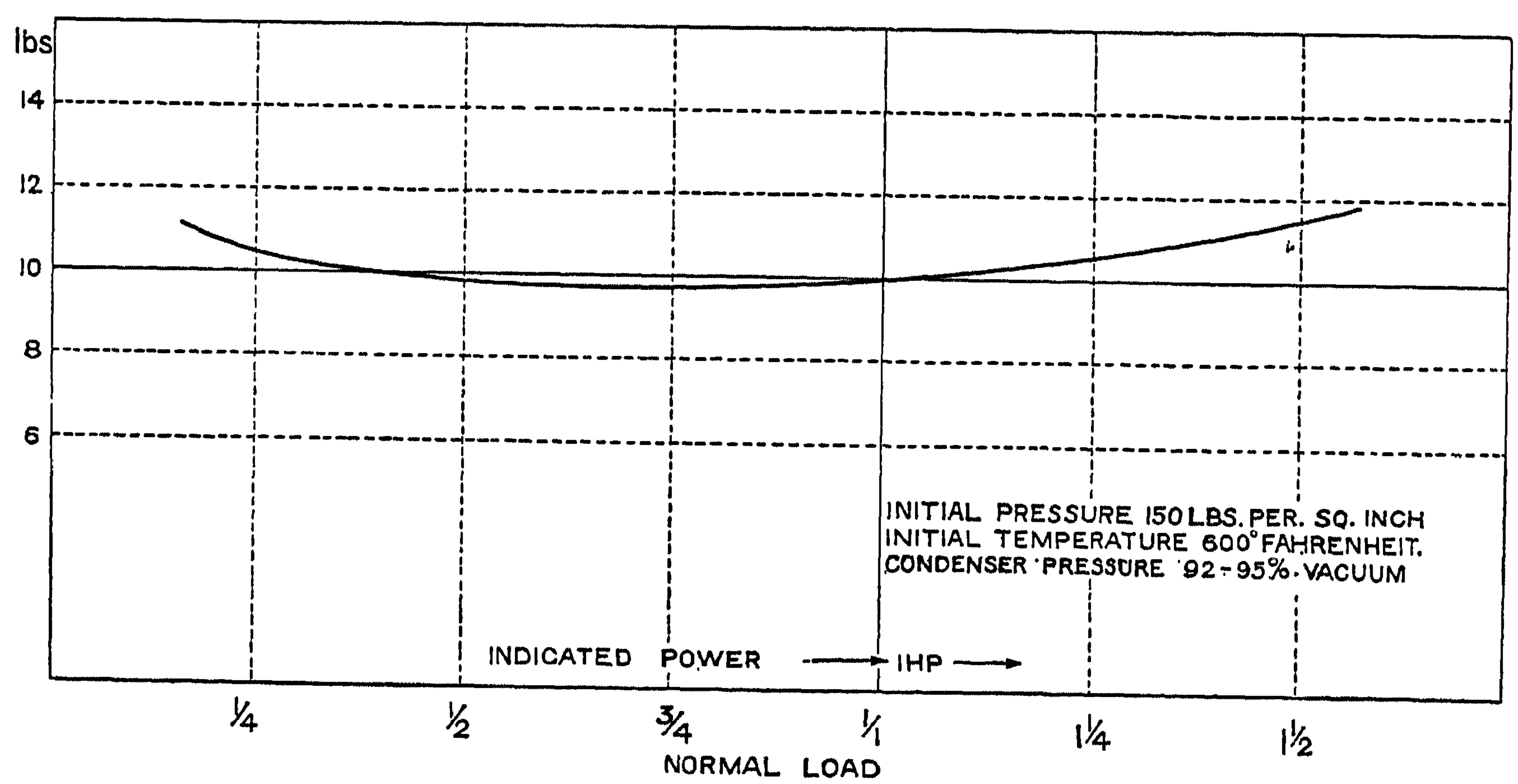


Fig. 8.—Steam Consumption per i.h.p. hour of a Sulzer Engine

From the figure it will be seen that in cases where an engine may have to work underloaded for a considerable time, as when a new installation is being put down with the expectation of more plant being added later, this type

of engine is very suitable. It is only fair to remark that most steam consumption figures for Uniflow engines are based upon indicated horse-power, the consumption per brake horse-power not being apparently available, so that a comparison with other types, the performance of which has been frequently ascertained, is not easily made.

The adoption of this type has entailed little alteration in general design or of details, with the exception, of course, of the cylinder, piston, and the bedplate, which is generally made more massive to resist the higher initial steam load, owing to all the work being done in one cylinder. Two bearings are usually provided, and the overhung crank is abandoned in favour of the double-web type. Uniflow engines for industrial purposes have hitherto been of the horizontal type only.

The advantages in economy given by superheating have long been known, but it is only in comparatively recent years that it has come into general use for large stationary engines. This is due to the adoption of mineral cylinder oils which resist the effect of high temperature much more than the animal and vegetable oils formerly used, and which gave trouble by carbonization, causing cutting and wear in the valve-chests and cylinders.

The use of superheat has necessitated considerably more attention being given to the design of cylinders and valves, in order to minimize the distorting effect of high temperatures upon irregularly shaped castings. Simplicity and symmetry are the guiding considerations. For the same reason large rubbing surfaces are avoided. Corliss as well as piston valves of the usual positive-driven type have been to a very large extent replaced by the drop-valve type.

The full inlet pressure of the steam acts upon the back of the Corliss valve, and when in the closed position presses it against its seat with great force. Considerable power is therefore required to drive it, and even when its lubrication is specially designed, the risk of cutting the faces is very great. The piston-valve, whilst being perfectly balanced against steam pressure, also has a large contact surface, and unless the working clearance be made ample, especially in the case of solid valves, it is always possible that a little distortion may give rise to similar trouble, whilst the increased clearance causes steam leakage. When rings are used in order to secure tightness, carbonization of the oil may cause them to become fixed in their grooves, and cutting, followed by seizing, may occur.

With the drop valve this risk is entirely removed, and tightness, the essential quality of a valve, is obtained without risk of seizing. The valve is almost perfectly balanced, and, owing to the spindle being of small diameter and the stuffing-box of special design, it is almost frictionless. The power required to operate the valves and gear is therefore negligible. The position of the valves with regard to the cylinder inlet is not complicated by consideration of other points in design, such as the position of the bearings and eccentrics on the shaft, and the ports can therefore be extremely short, reducing volumetric clearance to a minimum. This is also a good feature of Corliss valves. Sometimes piston drop-valves are used, working

in a liner provided with ports, and this type of valve is perfectly balanced.

Cylinders.—In the cases where Corliss or drop valves are used with the ordinary type of engine exhausting at the ends, there is one steam valve above and one exhaust valve below, at each end of the cylinder. This position of the exhaust valve ensures that the cylinders are completely drained of water during the exhaust strokes. The ports are kept as short as possible to reduce the clearance volume. The use of separate valves and passages for steam and exhaust avoids to some extent the initial condensation caused by the inlet steam coming first into contact with surfaces that have just been cooled down by being swept by the cooler exhaust steam. This feature, together with the better drainage, accounts for the superior economy of this type of engine when compared with the slide-valve type, in which the valves are placed on the side or even on the top of the cylinders, and the same ports are used for both live and exhaust steam, their position making them useless for drainage.

The two steam-valve chests are usually connected by a longitudinal passage on the top of the cylinder, the inlet branch being at the centre of the length of the cylinder. This construction is avoided in the case of the exhaust, as it would cause a considerable portion of the lower part of the cylinder barrel to be jacketed by exhaust steam, a condition which would not be good for economy. When a liner is used, this objection is not so great. The steam exhausts through the feet of the cylinder, a pipe in the case of Corliss engines being bolted to the under side of the feet and the exhaust taken off this pipe at the centre, or where convenient. The top of the foundation is, of course, suitably cut away to accommodate this pipe. For convenience in manufacture, some makers prefer to build up the cylinder in three castings. The barrel is perfectly plain, and is bolted between the two ends which contain the valves and to which the covers are secured, the exhaust escaping through the feet, which are incorporated with the end castings. A separate pipe or casting is used for the steam supply to the valve chests, splayed out at the ends to suit the slit-like openings above the valves, and having the steam branch or facing in the centre of its length.

In the case of drop-valve engines a liner is nearly always provided, and the valve chests and steam passages are cast with the cylinder.

The steam speeds in the ports are about 100 to 120 ft. per second for the inlet steam, and 80 to 100 ft. per second for the exhaust.

The thickness of cylinder barrels is fixed by the steam pressure, allowing a stress of about 1500 lb. per square inch in the metal, with an allowance for irregularity in thickness and for reboring, but in the case of large low-pressure cylinders, it is often decided by considerations of casting, a finished thickness of $1\frac{1}{2}$ in. for cylinders of 40 in. diameter and of 2 in. for 60 in. diameter being common.

The covers are always of the deep cellular design, the thickness of the metal being about 0·7 of that of the cylinders. The metal in the valve chests may have the same thickness.

The cylinder studs should have a diameter such that the stress at the

bottom of the thread does not exceed 3500 lb. per square inch for small studs, and 6000 to 6500 lb. per square inch for large studs 1½ in. diameter and over. The steam pressure should be taken as acting on a diameter equal to the pitch circle. The pitch of the studs should be from 3½ to 4 times the diameter of the stud when small, and 4½ to 5 times the diameter for large studs for high pressures. These figures may be 4¾ and 7 for low pressures. The thickness of the flanges may be equal to the diameter of

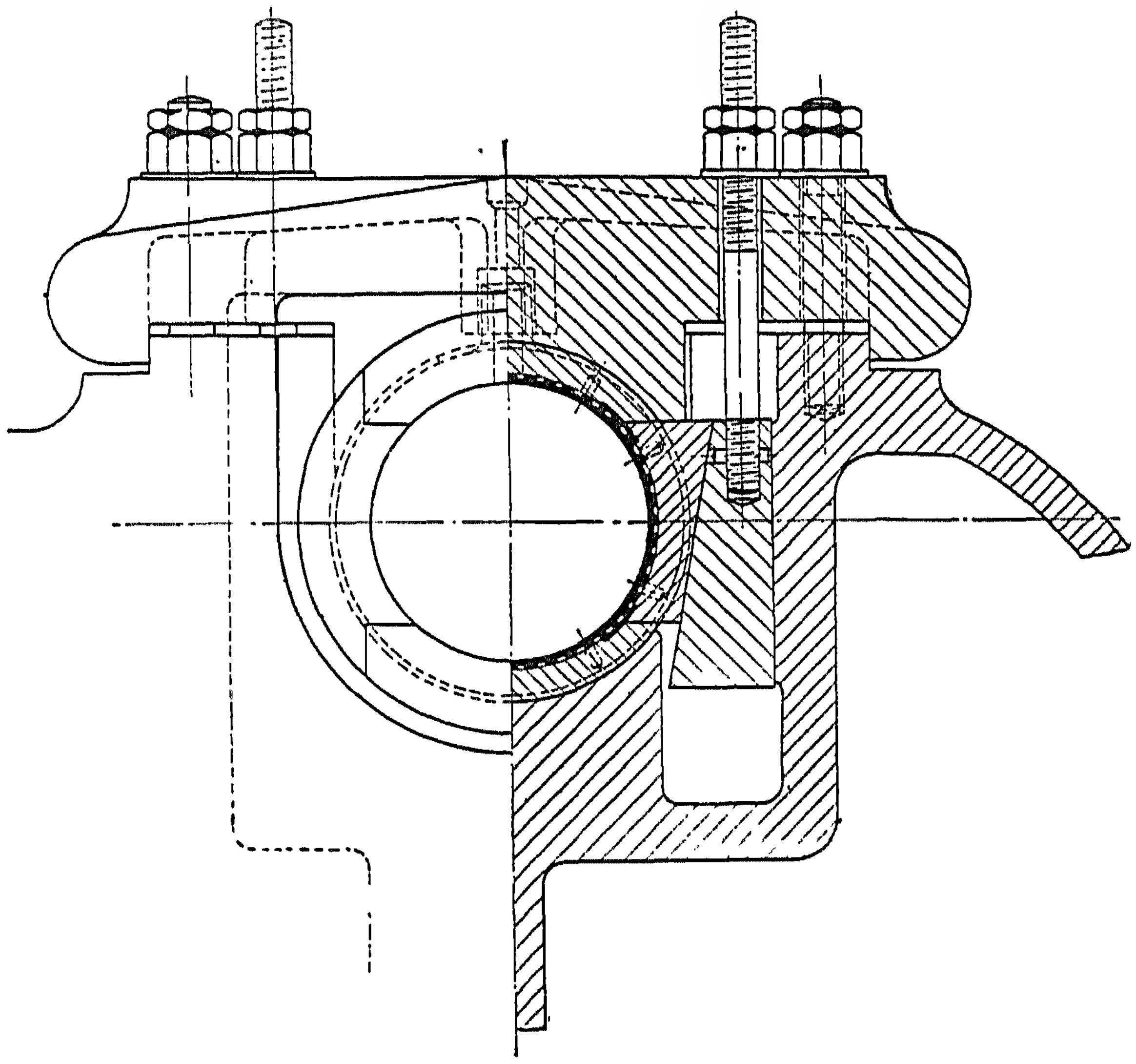


Fig. 9.—Design of Robey Main Bearing

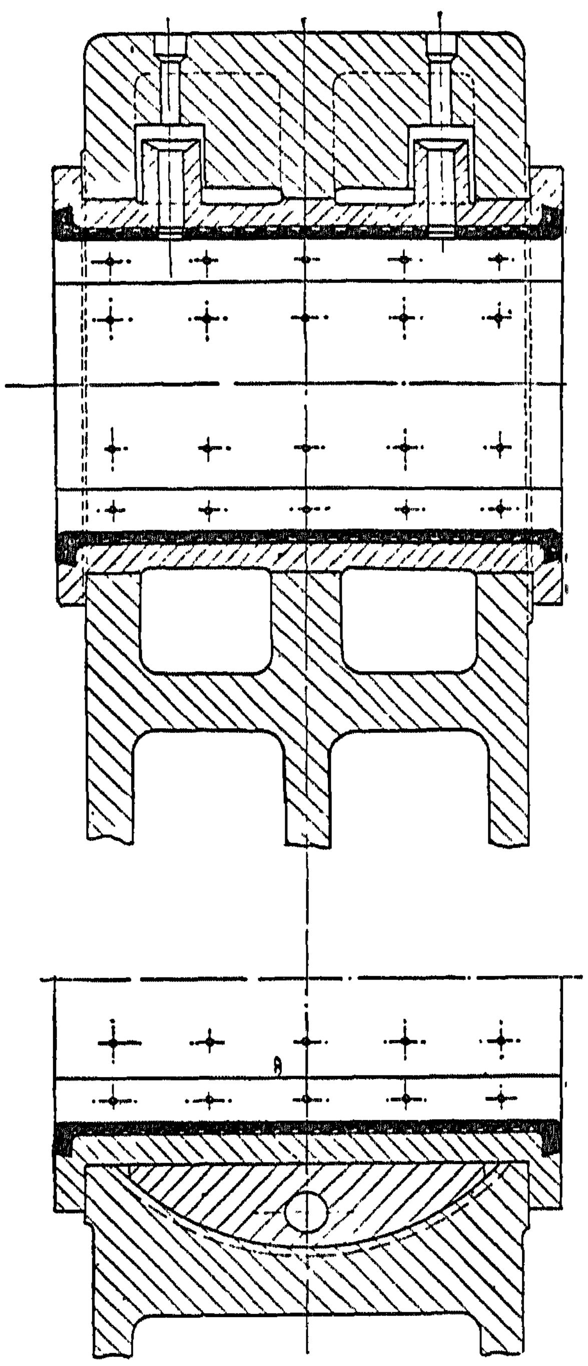


Fig. 10

the stud plus ⅜ to ½ in., and the width of flange on cylinder should be from 3 to 3½ times the diameter of the stud.

Bedplates.—For many years a common design of frame for engines with overhung cranks was of the "bayonet" girder type. The member carrying the main bearing was offset from the line of action of the steam pressure on the piston, bending stresses being thus set up which tended to reduce the rigidity of the structure. As steam pressures increased, and more especially since the development of the Uniflow engine with its high initial piston load, the double-sweep type of crankshaft with two bearings has been adopted, and the frame is made symmetrical about the centre line. This permits the metal to be placed more favourably to resist the load, and gives great increase in strength and rigidity.

The frame is bolted direct to the cylinder by a heavy flange, which is connected to the cylindrical part of the frame by a sweeping curve. The guides are usually of the bored type, or if flat, are loose and rest upon a bored seat. The forked part of the frame carries the main bearings, and is continued well back along the cylindrical part on both sides. In large engines these parts are sometimes separate castings and are joined together by a heavy flange connection. The end of the frame projects well beyond the bearings and rests on the foundations along the whole length, so that rigidity in all directions is well secured.

The main bearings are usually of the four-part type with wedge adjustment and are lined with white metal. Messrs. Robey & Co., of Lincoln, make a special design in which the back of the wedge is circular, or forms

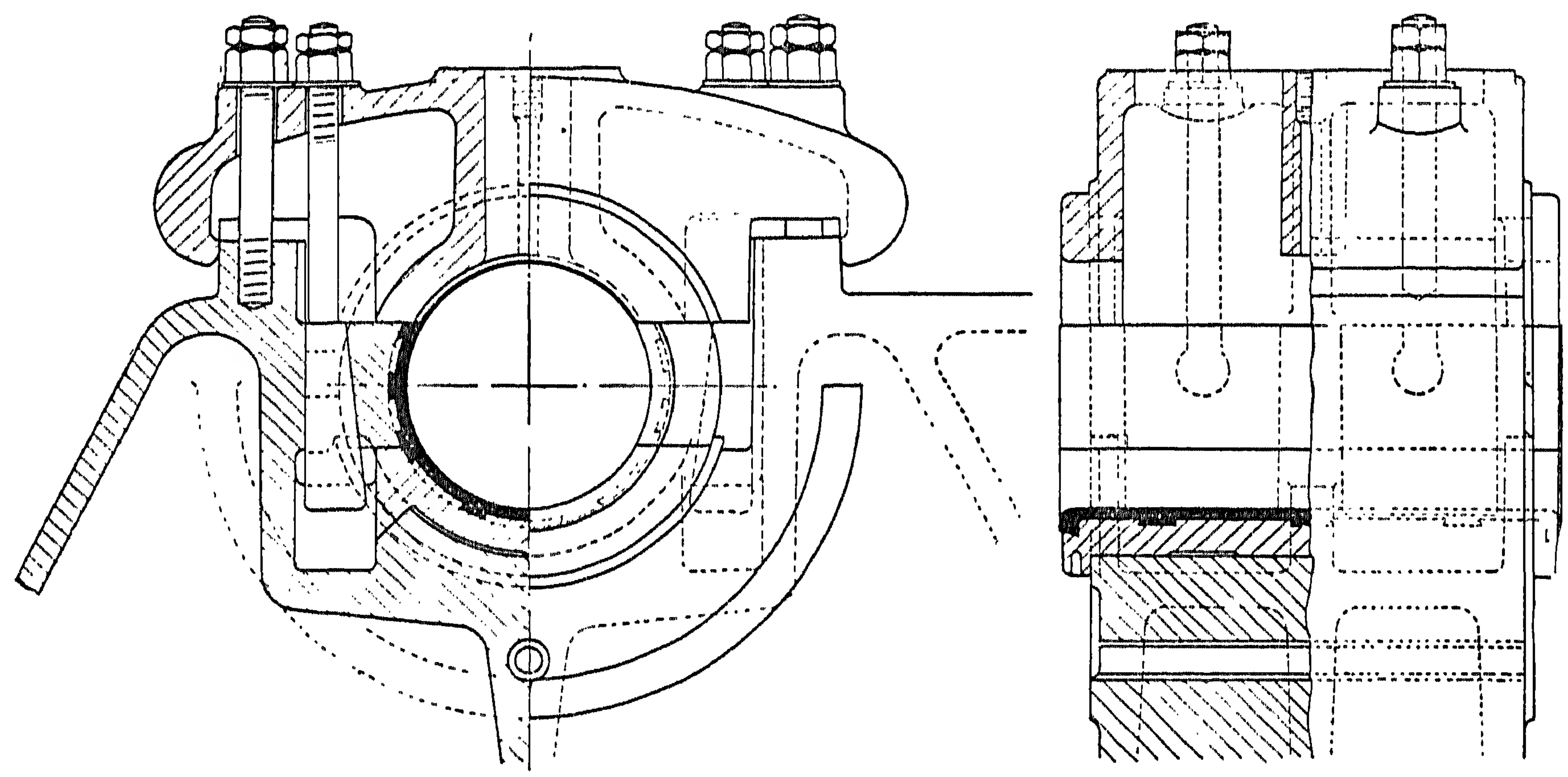


Fig. 11.—A Typical Design of Main Bearing

part of a cylinder, the inner side being, of course, inclined. This construction ensures an equable adjustment throughout the width of the wedge. This design is shown in figs. 9 and 10. A usual design is illustrated in fig. 11. The bearings are of cast iron lined with white metal as shown.

The pressure on the main bearings should be not more than 200 to 250 lb. per square inch due to the combined dead load and the steam pressure.

The thickness of metal in the frames is usually based upon manufacturing considerations, and the necessity for securing stiffness, but no part in tension should be stressed to a higher figure than 600 to 800 lb. per square inch.

The bolts or studs attaching the frame to the cylinder are usually greater in diameter than the cylinder cover studs, and a stress of 3000 to 4000 lb. per square inch at the bottom of the thread is allowed.

Piston-rod.—The design of the piston-rod is on similar lines to that in other classes of engine. A taper part of 1 in 4, and a parallel part with screw and nut, forms the attachment to the piston. Some makers adopt a taper of 1 in 3 on the diameter with a collar on the rod which really takes the thrust, the cone merely facilitating the withdrawal of the rod. A stress

of 5000 to 5500 lb. per square inch is allowed at the bottom of the thread, and a stress from 3000 to 3500 lb. per square inch in the body. At the crosshead end the attachment usually consists of a parallel part secured by a cotter, with the end of the rod butting against the bottom of the hole in the crosshead. Some makers adopt a cone of 1 in 3, as in the case of the piston end, but the end of the rod is the real driving part, the cone serving the purpose mentioned before.

The tensile stress on the section through the cotter hole should not exceed 5000 lb. per square inch, and this to a large extent fixes the diameter of the rod in the body. The cotter may have a thickness equal to not more than one-fourth the diameter of the rod, and a depth equal to the diameter, or even 1·2 times the diameter. The cotter is in double shear, and the stress may be 5000 to 5500 lb. per square inch. The bending stress on the

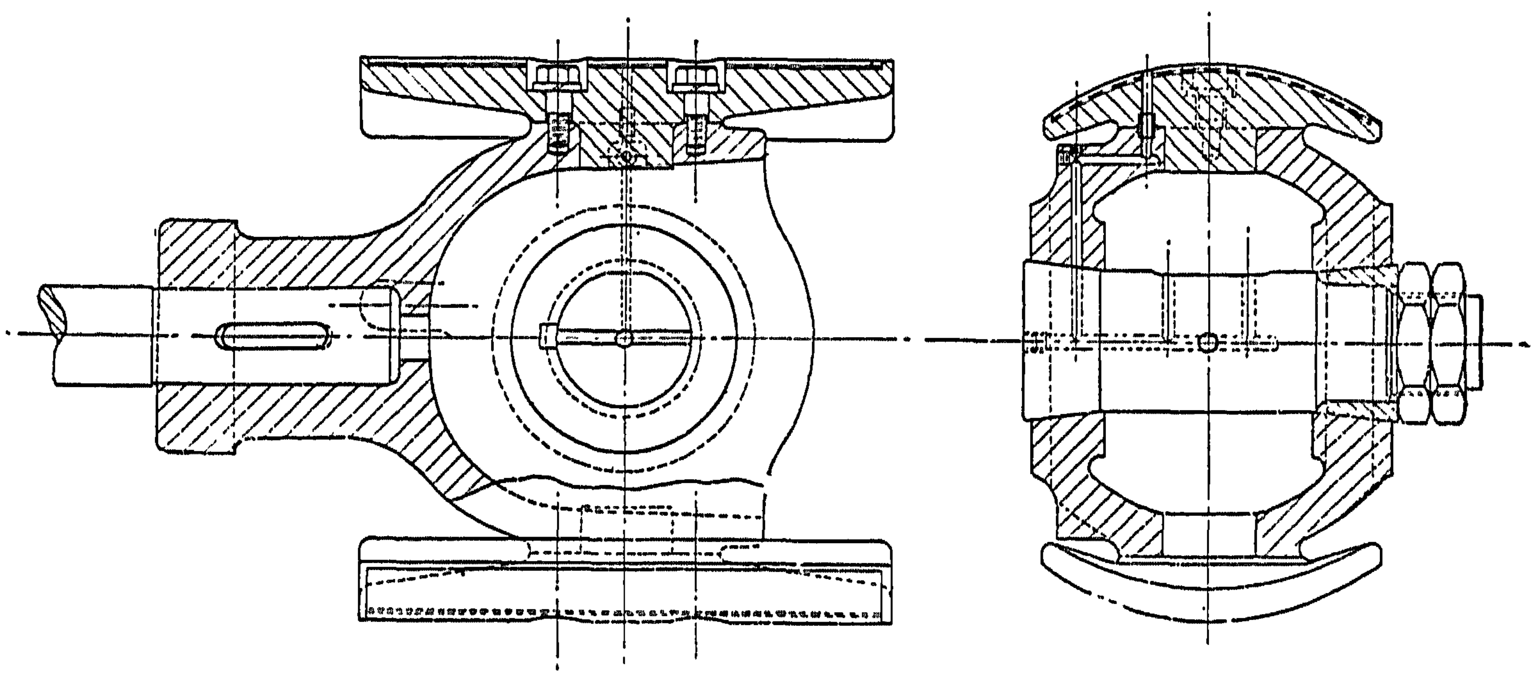


Fig. 12.—Design of Robey Crosshead

cotter should be checked, and, assuming a distributed load, the bending moment is $Pd/8$, where P lb. is the piston load and d in. is the diameter of the rod. The stress is in one direction, and should be about 8000 lb. per square inch. The stresses in the piston-rod, &c., are taken on the basis of the maximum effective load on the piston, due to the difference in pressure on the two sides. The cotter may have a taper of $\frac{5}{16}$ in. per foot, or about 1 in 40.

Some makers give the piston-rods an initial camber in a hydraulic press, just sufficient to overcome the weight of the piston.

Crossheads.—There are many varieties of this member. Bored guides are almost universal, and the slippers are of simple design usually of cast iron, but cast steel is sometimes used. The methods of attaching the piston-rod, and the designs of connecting-rod end, vary greatly, and these practically decide the type of crosshead. A good and reliable design is similar to that adopted in marine practice, where a forked connecting-rod end is used, with fixed gudgeons on the crosshead, and the piston-rod is fixed to the body by a nut and cone. The body consists of a cubical block of steel, to which the slippers are attached. A design used by Messrs. Robey & Co. is shown in fig. 12. This is for a solid-ended rod. The pin is fixed by a taper at one end, and a tapered split collar at the other,

both tapers being towards the centre line. The split collar is forced into a conical hole in the side of the crosshead by a nut on the end of the pin, thus tightly gripping the parallel part of the latter.

The pressure allowed upon the bearing surface of the gudgeons or pin may be from 1000 to 1200 lb. per square inch. The maximum pressure on the guides due to the obliquity of the connecting-rod may be 50 to 60 lb. per square inch.

Connecting-rods.—The marine pattern is often used, especially for the crank-pin end, but the rod with solid ends is common. Fig. 13 shows a rod by Messrs. Robey & Co. suitable for the crosshead illustrated

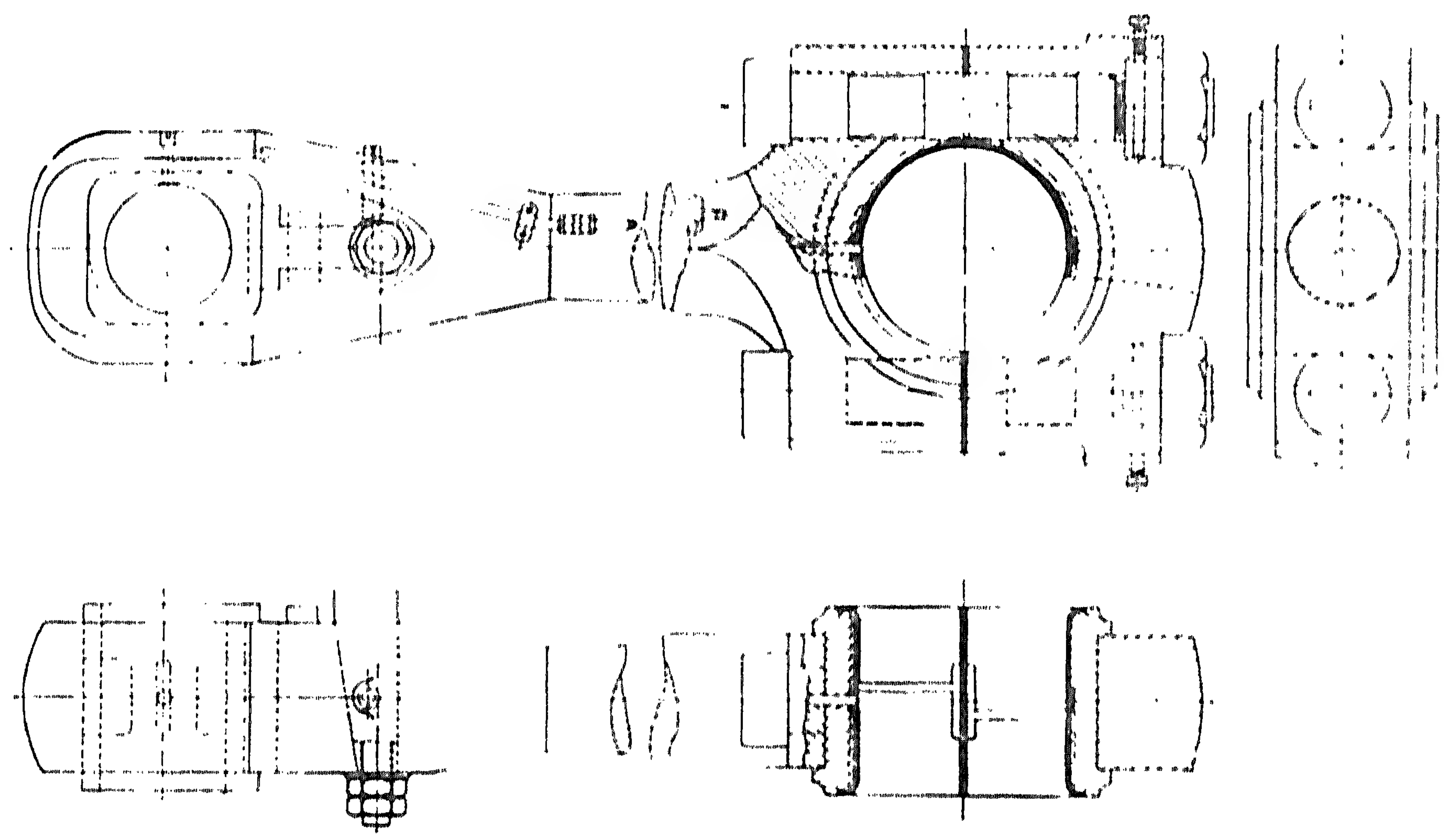


Fig. 13. Robey Connecting-rod

by fig. 12. The brasses at the crosshead end are adjusted by a wedge and screw.

Crank and Crankshaft. The journal of the crankshaft is subjected to both twisting and bending. In overhung cranks the twisting moment does not exceed *piston load × crank radius*, and the bending moment is taken as *piston load × distance between centre of crank pin and the centre of journal.* The greatest twisting moment occurs when the crank and connecting-rod are at right angles or thereabouts, but the greatest bending moment will occur when the steam load is greatest. The equivalent twisting moment should be taken for several positions to find its maximum, either from an actual or from an assumed indicator diagram, and the stress calculated from that maximum. The formula for the equivalent twisting moment is $T_e = M + \sqrt{M^2 + T^2}$, where M is the bending moment and T the twisting moment, taken as explained above. The maximum shearing stress is then $f = T_e/0{\cdot}196d^3$, where d is the diameter of the journal. A stress of 8000 to 9000 lb. per square inch may be allowed. Sometimes the equivalent bending moment is used, and is found from $M_e = \frac{1}{2}M + \frac{1}{2}\sqrt{M^2 + T^2}$,

where M and T are the bending and twisting moments as before. The stress $f = M_e/0{\cdot}0982d^3$, but in this case the stress figured is a tensile stress.

In the case of overhung cranks the crank-pin is fixed in the crank cheek. The crank-pin is designed for bearing pressure which should not exceed 600 lb. per square inch. The length may be $1\frac{1}{4}$ times the diameter. The part fixed in the crank is a little larger in diameter than the pin and is parallel, a fillet being left at the change of sections. The pin is fixed in by shrinkage.

The crank is usually a plain mild-steel slab, having a thickness equal to diameter of shaft $\times$ 0·6. The thickness of metal round the hole for the pin may be equal to the radius of the latter, and the metal round the hole for the shaft may be equal to the radius of the hole $\times$ 0·85 to 0·9.

The crank seat on the shaft is slightly greater in diameter than the journal, to which it is joined by a fillet. The crank is fixed on by shrinkage, a round key, half in each part, being afterwards driven in. The key may have a diameter of about one-eighth of the diameter of the crank-shaft.

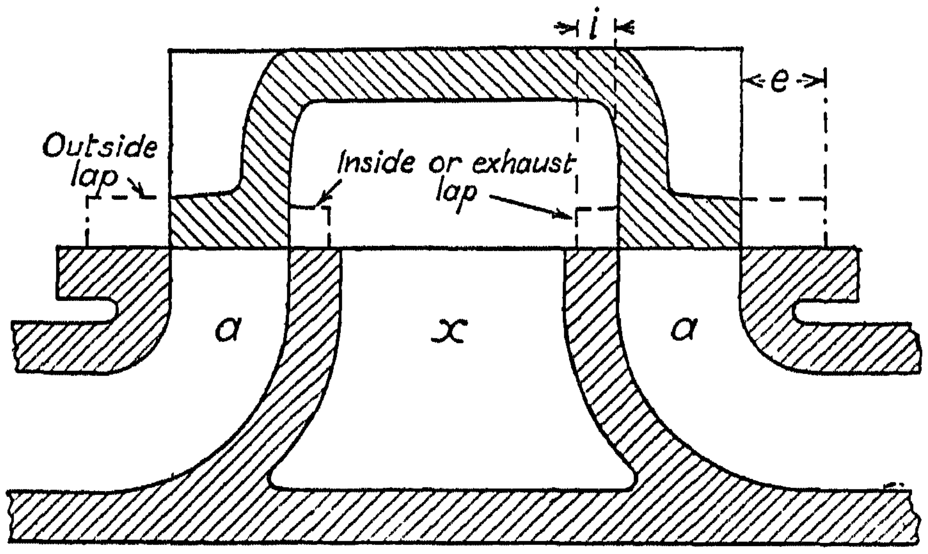


Fig. 14.—Slide Valve

The above are usual proportions for mill engines, running under ordinary conditions, but it would be well to check the bending stresses on the pin, also in the crank in both planes, for any unusual conditions of steam pressure, allowing a stress of 8000 lb. per square inch.

Valves and Valve Gear.—The slide valve shown in fig. 14 is the most common type, and when used with low or moderate pressures, is quite satisfactory. With high pressures, or with superheat, the faces of the valve and cylinder are likely to be cut. When large, this type of valve requires considerable power to drive it. The valve, as shaded, exactly covers the three ports. If it projects outside, it has *outside* or *steam lap*, the amount of which is measured by e in the figure; if it projects inside as well, it has *inside* or *exhaust lap*, marked i.

In the first place, a valve with no lap will be considered. In the arrangement shown in fig. 14, there are two steam ports a with the exhaust port x in the centre. The flat face of the valve is equal in length to the total width of the three ports, in addition to the width of the bars of metal between the exhaust port and the steam ports. The valve is shown in its central position. Assuming the valve to be moved to the right, the left-hand end of the cylinder would be put into communication with the steam chest, in which the valve works, and the right-hand end with the exhaust passage, and the piston would be forced from left to right. While the piston was moving throughout its stroke, the valve would have completely opened the port, and then moved back to close it, being in the full open position

at the same time that the piston would be at mid-stroke, neglecting the obliquity of the connecting-rod, so that by the time the piston had arrived at the right-hand end of the cylinder, the valve would have returned to its central position. It would continue to move to the left, and would thus open the right-hand port to steam and the left-hand port to exhaust, and the piston would move from right to left, and so on continuously. It will be seen, therefore, that the valve travels from mid-position by a distance equal to the port opening, and that when it is in mid-position the piston is at one end or other of its stroke, and vice versa.

A little consideration will show that the crank or eccentric driving the valve must lead the main crank by 90° in the direction of rotation, as shown

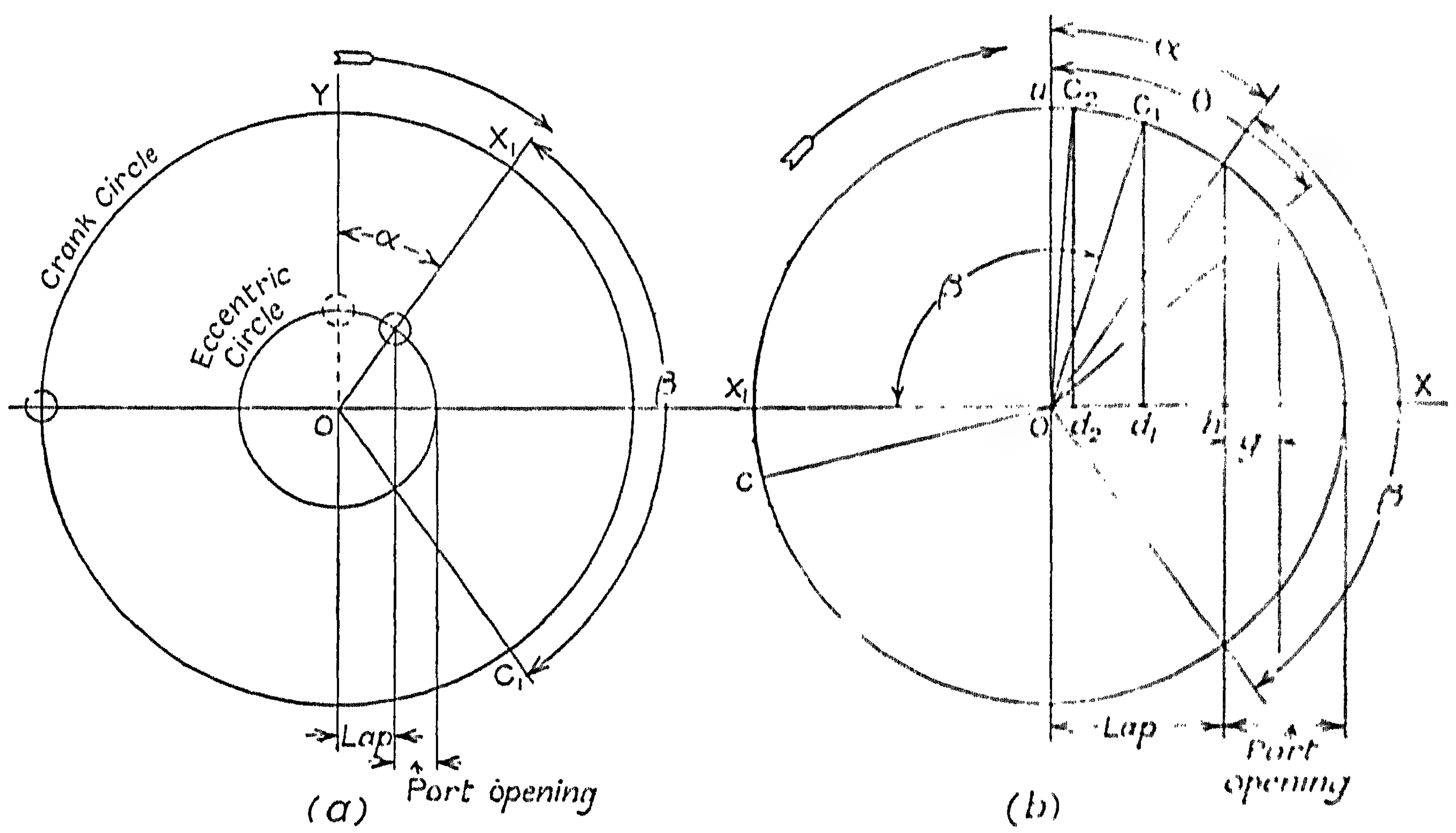


Fig. 15.—Valve Diagrams

by the arrow (fig. 15 *a*), while the time occupied for the complete opening and closing of the port, or, in other words, the *period of admission*, is just equal to the time taken by the piston to make one stroke, or while the crank is passing through 180°, the steam being admitted during the whole of that period. This is, of course, wasteful, as the expansive force of the steam is not used. To obtain expansive working the steam must be "cut off" from the cylinder at some point before the piston has completed its stroke. This is done very simply.

The valve is made longer at each end so that it overlaps the outer edges of the steam ports, as shown by the dotted lines in fig. 14, and in order that the port shall be opened at the same instant as before, that is, just when the piston is commencing its stroke, the eccentric must be moved forward on the shaft in the direction of rotation through an angle α which would give a movement *equal to* the lap. This is called the *angle of advance*, and is obtained by setting off the lap from O (fig. 15 *a*), and drawing upward the perpendicular to cut the valve circle. The period of opening and closing, or of admission, is now the angle β, found by producing

the perpendicular downwards to cut the valve circle. This angle is less than 180°, the period for the valve without lap.

In fig. 15 *b* the eccentric circle is drawn to an enlarged scale. Suppose ϕ is the angle that the eccentric crank makes with the line of stroke of the engine and ψ the angle that the main crank makes with the same initial line, then

$$\phi = 90 + \alpha + \psi,$$

where α is the angle of advance, i.e. the angle YOX_1 of fig. 15 *a*. Hence

$$\phi - (90 + \alpha) = \psi,$$

i.e. if we rotate the valve diagram, giving the position of the eccentric, backwards through an angle $(90 + \alpha)$, the new diagram gives the positions of the main crank at the corresponding valve positions. Now rotate the valve diagram (fig. 15 *a*) so that OX_1 is rotated backwards through an angle $(90 + \alpha)$. The figure now obtained is shown in fig. 15 *b*, where X_1 gives the direction of the *main crank* at the valve positions corresponding to X_1 in fig. 15 *a*, i.e. OX_1 in fig. 15 *b* gives the direction of the crank at admission. Suppose now the angle X_1OC_1 is set off equal to β, then OC_1 in fig. 15 *b* gives the direction of the crank corresponding to valve-position OC_1 in fig. 15 *a*. Now, if the connecting-rod is very long, we can drop C_1d_1 perpendicular to OX, and Od_1 gives the displacement of the piston from its mean position.

In order to assist in bringing the moving parts to rest, especially in quick-running engines, some form of "cushioning" must be provided. Such a steam cushion could obviously be provided if the connection from the cylinder space at the back of the piston to the exhaust pipe were closed a little before the piston had reached the dead centre on the exhaust stroke. All that is necessary to bring about this state of things is to provide *inside* or *exhaust lap* on the valve as is shown in fig. 14, marked *i*.

It is also desirable that the valve admitting "live steam" to the back of the piston should be open to a certain extent when the piston crosses the dead centre. This effect is brought about by slightly increasing the angle of advance, so that the valve is displaced a little more than it normally would be, when the piston is at the dead centre. The effect of this is, of course, to open the port slightly before the piston has reached the dead centre, so that *pre-admission* occurs. The amount by which the port is open when the crank is actually on the dead centre is called the *lead*, and it varies in amount with the type and speed of the engine. A very simple relation connects the quantities, outside lap, lead, throw of valve, and angle of advance. It is

$$e + l = r \sin\theta,$$

where e is the outside lap, l the lead, r the throw of the eccentric, and θ the angle of advance, that is, the normal angle of advance α plus the increment which must be provided to secure the lead. The total angle of advance θ

is found by setting off the lead g from h and drawing a perpendicular to the valve circle, giving the position of the eccentric with the crank at X_1.

When the main-crank angle is ϕ, the eccentric-crank angle is $\phi + (90 + \theta)$,

$$\text{i.e.} \qquad \phi + 90 + a + (\theta - a),$$

therefore the main crank is at $-(\theta - a)$, putting $\phi = -(\theta - a)$, the eccentric-crank angle is

$$-(\theta - a) + 90 + a + (\theta - a), \text{ i.e. } 90 + a,$$

and, as already seen, admission begins when the eccentric-crank angle is $(90 + a)$. Hence to find the point of admission set off in fig. 15 *b*, $X_1OC = (\theta - a)$, drawing the angle in the negative direction.

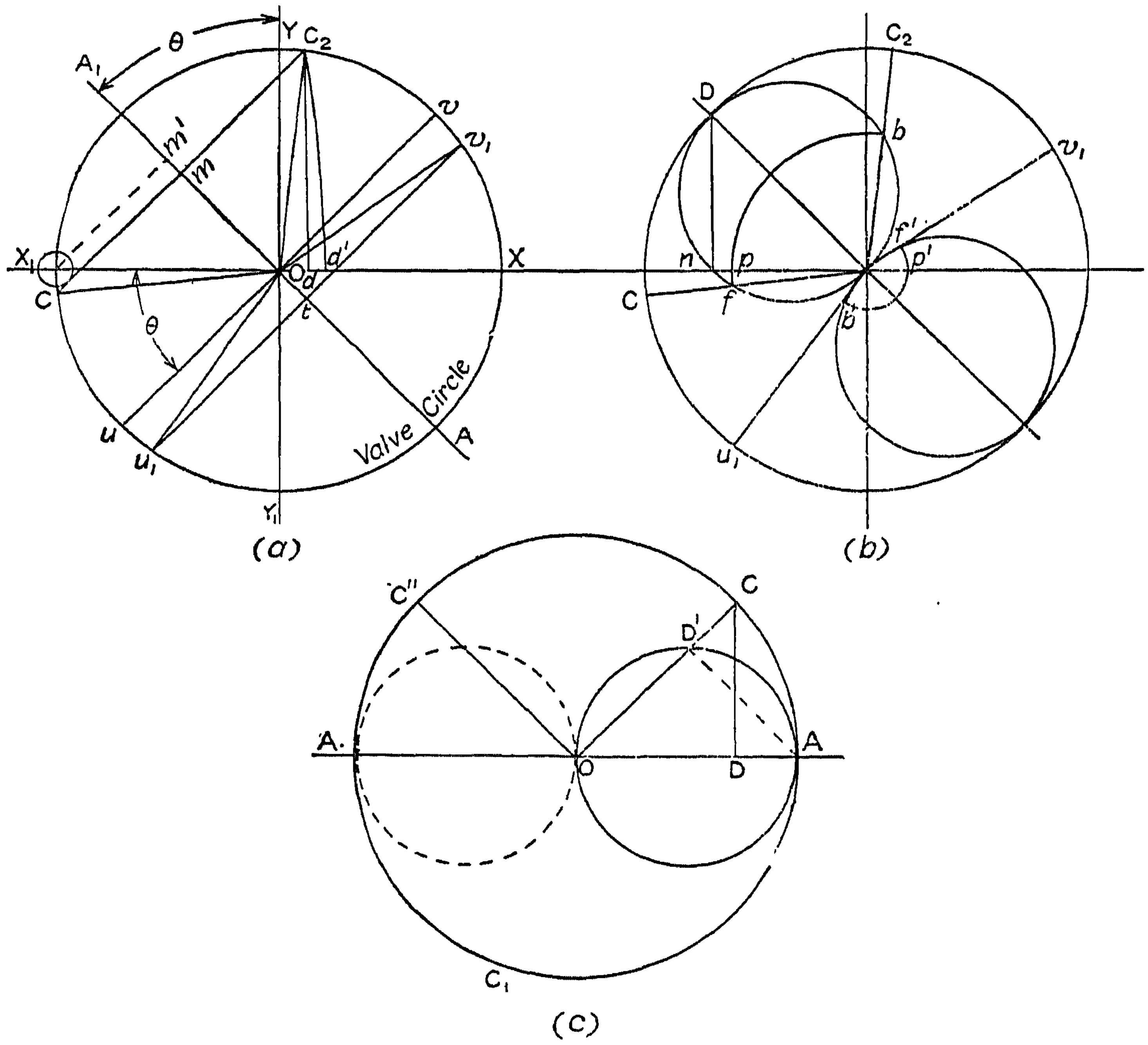


Fig. 16.—*a*, Reulaux Valve Diagram. *b*, Zeuner Valve Diagram. *c*, Zeuner Circles

Similarly when ϕ is $\beta - (\theta - a)$, the eccentric angle is $\beta - (\theta - a) + 90 + a + (\theta - a)$, i.e. $90 + (a + \beta)$, i.e. from fig. 15 *b*. C_2 corresponds to the position of the main crank at cut-off, then angle C_1OC_2 is $(\theta - a)$.

The important information wanted is the position of the main crank or the piston at the important events of *admission*, *cut-off*, *release*, and *compression*. Since the valve diagram leads the crank diagram by $(90 + \theta)$, all that is now required is to rotate the valve diagram backwards through $(90 + \theta)$, and read it as a crank diagram. This is done in fig. 16 *a*, where O*u* is the line O*u* of fig. 15 *b* rotated backwards through $(90 + \theta)$, i.e. O*u* is drawn at θ on negative side of OX_1. Through O draw AA′ perpendicular to O*u*.

Set off Om equal to the outside lap and draw CC_2 through m perpendicular to AA′. Then C is the main-crank position at admission and C_2 of the cut-off. Likewise set off Ot on the opposite side from Om equal to the inside lap, then v_1 is the release point and u_1 the compression point.

The lead is the amount the valve is open when the main crank is at dead-centre, i.e. at X_1 *considering the cover-side of the cylinder.* The valve displacement when the crank is at X_1 is Om'. The valve lap is Om, therefore the opening is Om' less Om, i.e. $m'm$, i.e. the radius of the *lead circle at* X_1.

If the given data is the point of cut-off, the valve travel, and the lead, then the known dimensions are:

(*a*) the radius of the valve circle, i.e. OC;
(*b*) the radius of the lead circle at X_1, i.e. l;
(*c*) the position of C_2.

The construction is then: draw a circle of radius equal to half of the valve travel and rule perpendicular lines XX′ and YY′. With X_1 as centre draw a circle of radius l. Draw a tangent through C_2 to this circle intersecting the valve circle at C. Draw AA′ through O perpendicular to CC_2 and bisecting it. The angle A′OY is the angle of *advance* θ, Om the *outside lap*. To find the inside lap the point of release or compression would have to be prescribed.

Obliquity of Connecting-rod.—If the connecting-rod were infinitely long, the piston-displacement could be found, corresponding to the events—*admission*, *cut-off*, *release*, *compression*—by dropping perpendiculars from C, C_2, v_1 and u_1 (fig. 15 *a*) on to XX_1. Thus Od is the displacement of the piston from its mean position at cut-off when the connecting-rod is very long relative to the main-crank throw.

If the connecting-rod is not very long a length is taken on a pair of compasses representing the length of the connecting-rod on the same scale as the radius of the valve circle represents the main-crank throw. Taking a centre on XX_1 produced, strike an arc through any representative point, say C_2. The intersection of the arc with XX_1 marks d'—the actual point of admission, so that Od' is now the true piston displacement allowing for the obliquity of the connecting-rod.

Zeuner's Diagram.—A very simple modification can be made by means of Zeuner's circles. Suppose CC_1 (fig. 16 *c*) is the valve circle and C a representative point on the circle, the angle AOC being the "phase" of the eccentric at C, i.e. ψ. With centre on AA′ and radius $\frac{1}{2}$OA′ draw the small circle shown. OD is the displacement of the valve from its mean position. But OD = OD′, for the triangles ODC and OD′A′ have equal angles and a side OA′ and OC equal. Hence if a radial line is drawn to C from O the intersect OD′ made by the small circle measures the valve displacement.

Obviously another circle can be added, shown dotted in fig. 16 *c*, to give the displacements radial lines such as OC″, and if these Zeuner circles are added together on the line AA′ of fig. 16 *a*, fig. 16 *b* is obtained.

Striking now the arc *fpb*, fig. 16 *b*, of radius equal to the outside lap and the arc *f'p'b'* of radius equal to the inside lap, the history of events may be traced very simply.

Thus at *f* the valve is just about to open, i.e. C is the position of the main crank at *admission*; *pn* is the opening of the valve at dead centre, i.e. *pn* is the *lead*; *b* corresponds to *cut-off*, i.e. C_2; and so on.

Crank Side of Piston.—If the engine is symmetrical, these diagrams can be used directly for the valve design for this side of the piston. They, of course, have to be rotated through 180° to give correct main-crank positions for the different events. If the engine is unsymmetrical a new set of diagrams will be required for the crank side of the piston in a double-acting engine.

In some cases when piston valves are used, steam is admitted at the inner edges and exhausted at the ends. The eccentric is then placed at an angle of 180° with an eccentric for driving a valve with " outside " admission, the crank thus leading in the direction of rotation. All investigations in designing can be made on the bases of outside admission, the eccentric being keyed to the shaft in the position mentioned.

Valves, Valve Gear, and Eccentrics.—As stated above, the steam and exhaust valves for large stationary engines are now nearly always of the drop-valve type, and a number of illustrative examples are given. In many cases the valves are operated by eccentrics mounted on a lay shaft driven by either bevel or spiral gear from the main shaft.

The high-pressure valves are worked through a device containing a member which is automatically put into gear with the lever lifting the valve spindle in order to open the valve at the proper time, but which is disengaged by its coming into contact with another member some time during the opening movement. The position of this member is regulated by the governor. The valve lever being thus released, the valve closes, partly by its own weight (for these valves always move vertically, and open upwards) and partly by the force of a spring acting upon the upper end of the valve spindle. The power developed by the engine is thus controlled by alteration to the cut-off between the limits of speed for which the governor is designed. Corliss valves are operated in the same way, the only difference being that the motion is angular instead of up and down. The variety of such " slip " or " trip " gear is endless, much ingenuity having been spent upon their design under the belief that a sharp cut-off greatly improves the economy.

The superior performance of engines fitted with this type of gear is the result mainly of a combination of other factors, such as reduced clearance volume, for the valves can be placed almost inside the cylinder; the use of separate ports for inlet and exhaust, which greatly reduces initial condensation; and the good drainage owing to the exhaust valves being placed at the bottom of the cylinders in horizontal engines. The valves also close much more tightly than either piston or slide valves.

The property of being able to vary the cut-off gives a ready means of meeting varying demands for power throughout a wide range, and this

method of regulation would on the whole cause less steam to be used compared with the regulation by the throttle, but the quick cut-off in itself has little effect upon economy. Latterly, drop valves have almost entirely displaced Corliss valves, and the former type only will be referred to. They require very little power to operate, and their construction allows a tight condition to be easily maintained by occasional regrinding on their seats.

An example of this type of valve with positive gear is given in fig. 17, which shows a design by Messrs. Musgrave & Co.

The variation in design of valve gears is very great, but the principles are the same throughout.

In the case of Uniflow engines the cut-off is necessarily very early, in order to expand high-pressure steam in one cylinder down to condenser pressure, and the whole movement of opening and closing the valve takes place during the time that the crank passes through a small arc only. The time available for the engagement of trips would be so short that gears of this class are not satisfactory. So-called positive gears are usually employed, an example of which, made by Messrs. Musgrave & Co., is shown in fig. 17, together with the closing spring, valve spindle, and the valve and seat. A dashpot is not required with this type of gear. A steel cam plate is fixed in a guide or piston attached to the valve spindle, and a roller, fixed in a groove in a bar sliding in guides in the cover and actuated by the eccentric, comes into contact with the cam face and lifts the valve, gradually at first, then quickly to its fully open position. This action is reversed on the return stroke of the slide bar, so that at first the valve closes rapidly, but later is lowered gradually and without shock on to its seat under the action of the

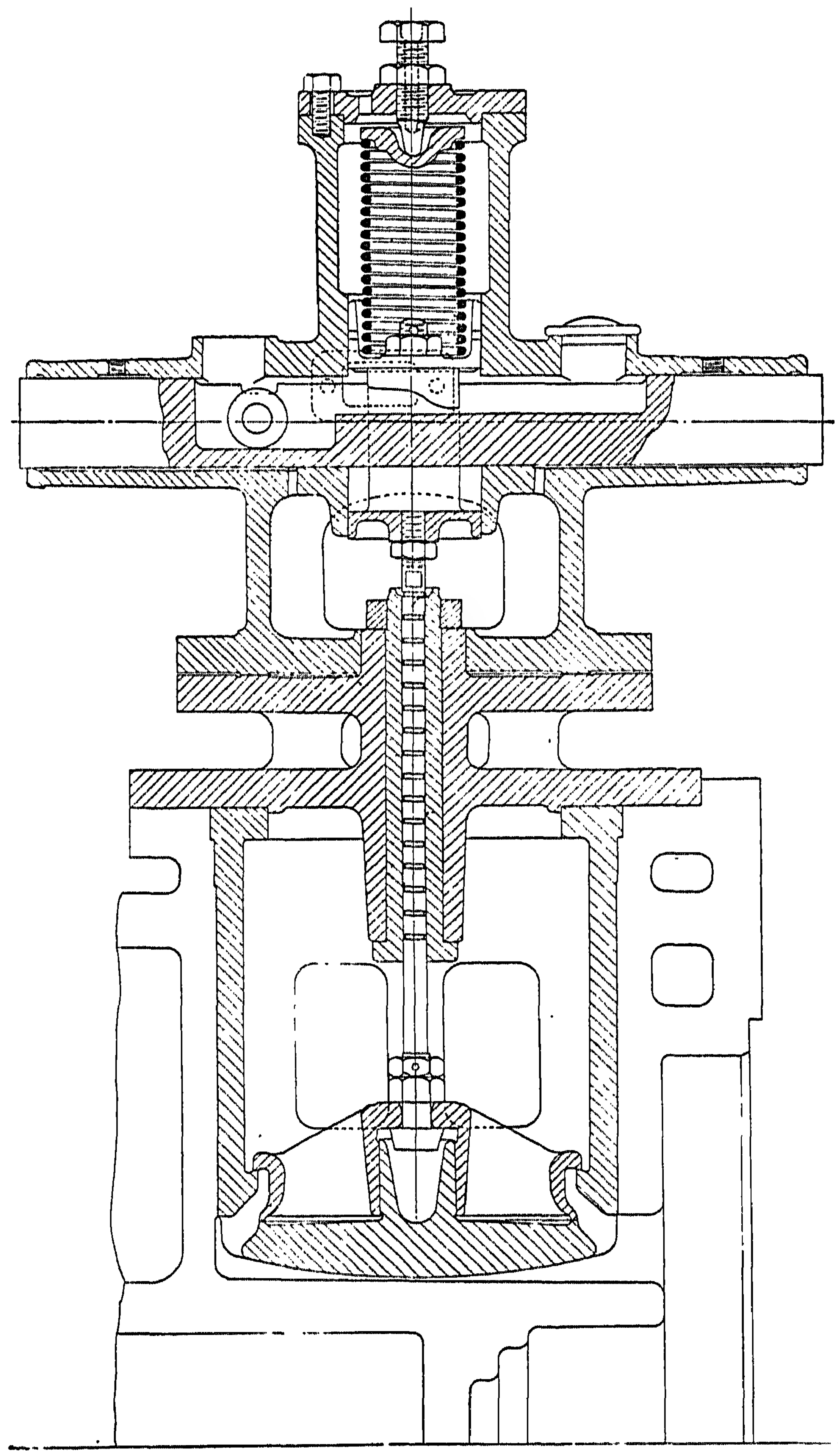


Fig. 17.—Drop Valve by Messrs. Musgrave & Co.

spring. The groove in which the roller is fixed contains oil for lubrication.

The valve is of cast iron of the usual double-beat type. The faces are flat and narrow, the lower face being of slightly less diameter than the upper, so that the valve is not in exact equilibrium. The valves also are of cast iron, and the distance between the two seats is made short, to reduce as much as possible the difference in axial expansion between the valve faces and the faces in the seat. The valve is guided in the centre by a projection of the seat. No stuffing-box is required. The valve spindle is provided with water grooves, and slides in a long bush fixed in the cover, leakage being thus prevented.

Governor.—The simple pendulum governor was invented by Watt, and consists essentially of an arm, suspended from a vertical spindle in such a way that it is free to move round the point of suspension in a vertical plane passing through the axis of the spindle. The free end carries a weight or ball. In practice two arms are used for obvious reasons.

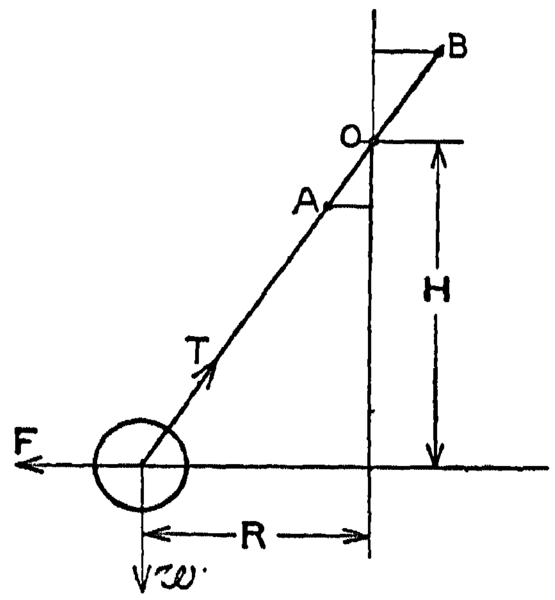


Fig. 18.—Simple Pendulum Governor (suspensor of governor arm).

The arm may be suspended in three ways: (*a*) from a point O situated in the vertical axis, (*b*) from a point A on the same side of axis as the weight, or (*c*) from a point B on the other side of axis. The point O is important. In the case (*a*), fig. 18, it is the point of suspension and is fixed. In case (*b*) it is the point of intersection of the centre line of the arm produced to the axis, and in case (*c*) it is the point of intersection of the centre line of the arm and the axis. With this latter arrangement the governor is said to have crossed arms. In the cases (*b*) and (*c*), O is not fixed, but varies with the angular position of the arms with regard to the axis. In the former case, O moves in the direction opposite to that of the weight, that is, when the ball moves outwards and upwards the point O descends. With crossed arms the point O moves in the same direction on the axis as the ball moves vertically.

The vertical distance H between the point O and the horizontal plane in which at any instant the balls revolve, is called the height of cone, and the fundamental fact, relating to the vertical gravity-controlled governor, is that H varies inversely as the square of the speed of revolution.

In fig. 18 the ball is in equilibrium under the action of the three forces, the weight w, the centrifugal force F, and the reaction or tension T along the arms. Taking moments about O, $\mathrm{FH} = w\mathrm{R}$, or $\mathrm{H} = \dfrac{w\mathrm{R}}{\mathrm{F}}$. But $\mathrm{F} = \dfrac{wv^2}{g\mathrm{R}}$ where v is in feet per second and $v^2 = 4\pi^2\mathrm{R}^2\mathrm{N}^2$. Therefore $\mathrm{H} = \dfrac{g}{4\pi^2\mathrm{N}^2} = \dfrac{0{\cdot}81}{\mathrm{N}^2}$, and thus varies inversely as the square of the revolutions. In the above expression R and H are in feet and N in revolutions per second.

If h is the height of cone in *inches*, and n is the speed of revolution per *minute*, then $h = \frac{35{,}200}{n^2}$, i.e. the height of cone is inversely proportional to the square of the speed.

For a change of speed from 65 to 75 r.p.m., the alteration in height of cone is $h_1 = 2{\cdot}1$ in., whereas for a speed change from 100 to 110 r.p.m., the alteration is only $h_2 = 0{\cdot}6$ in. This shows that such an arrangement is valueless for governing at high speeds.

Sensitiveness, Power, and Stability.—The meaning of these terms must be defined. Sensitiveness is the ratio of the total variation of speed to the mean speed $\frac{n_2 - n_1}{n}$. Any governor may be made as sensitive as we please by choosing its range of speed variation sufficiently small. The range of movement would then be small also, and would have to be multiplied up by levers or their equivalent in order to obtain the necessary range of movement in the gear controlling the admission of steam, with the result that the effort available would be barely sufficient to overcome friction alone. In comparing two governors it is necessary, therefore, to know not only the speed sensitiveness or variation, but also the amount of work available for external use stored in moving from the position of lowest speed to that of highest speed. In gravity-controlled governors the amount of stored work is obviously the product of the weight of the balls and the vertical distance through which they move with regard to the earth, and this is not necessarily the same as the change in H. When the arms are pivoted on the axis, this vertical distance is the same as the variation of H. In case (*b*) it is less, as the point O descends as the balls rise. In case (*c*) it is more, as the point O rises with the balls, but not at the same rate, and for this reason the cross-armed governor is a little more powerful than either of the other types.

The plain Watt governor could be made both as sensitive and as powerful as we please by making the arms very long, and using the lower range only of the movement of the balls. Thus, a governor running at a lower speed of 20 r.p.m. and an upper speed of 20·6 r.p.m., giving a speed variation of 3 per cent reckoned on the lower speed, would require heights of cone of 88 in. and 83 in. at the two extreme positions, respectively. Very heavy balls could be used, and the governor would be sensitive, powerful, and stable, but also impracticably cumbrous.

It has been shown that for all forms of the pendulum governor there is a definite height of cone and position of the balls for each speed. It is therefore impossible for the mechanism to alter its position without a change in the speed. This quality is called " stability ", and is necessary for good governing, entailing the condition that some variation in speed is unavoidable.

In the description of high-speed engine governors the quality of isochronism is referred to, and it is shown to be obtainable with certain arrangements of weights and springs, but quite useless. This quality may be obtained in the case of gravity-controlled governors in various ways, for instance, by causing the balls to move in a parabolic path co-axial with

the spindle. The parabola has the property that the subnormal h is constant (fig. 19). Therefore the height of cone would be constant. Only one speed would be possible, and the balls could take up any position within their range. This is a condition of neutral equilibrium.

The cross-armed governor is the simplest form of vertical gravity-controlled governor, in which the path of the balls may be made to approxi-

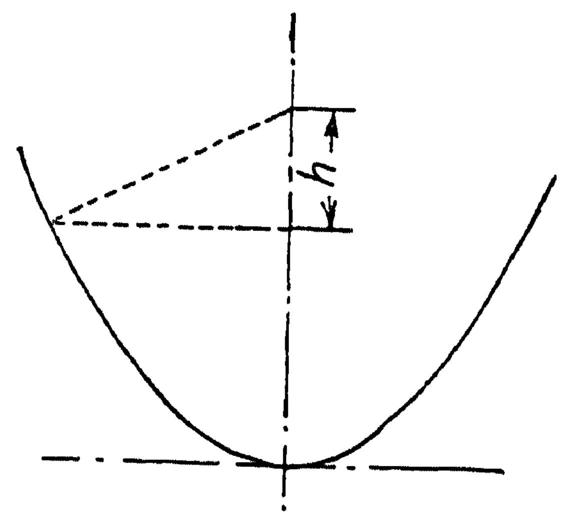


Fig. 19

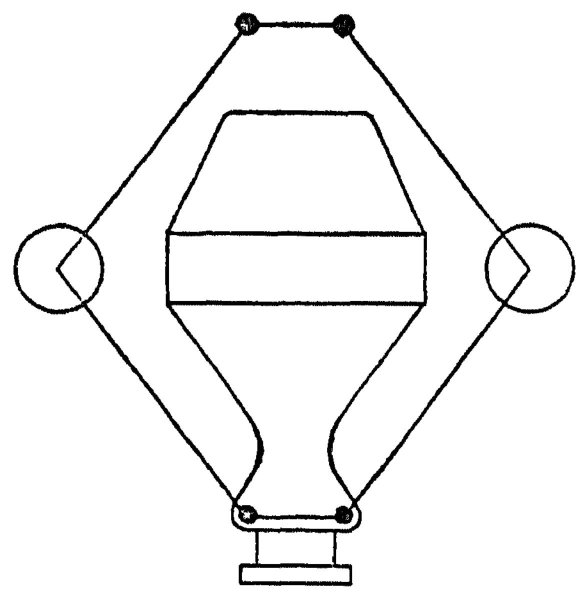


Fig. 20.—Porter Open-arm Governor

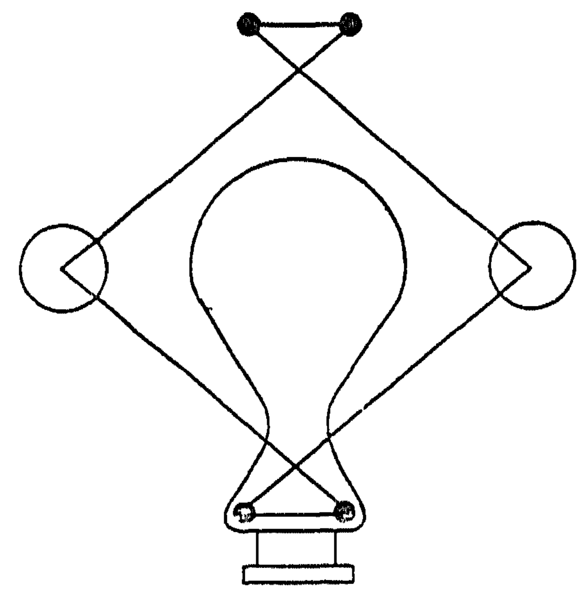


Fig. 21.—Porter Crossed-arm Governor

mate to parabolic arcs in the range of movement usual, but the point of suspension should not be situated too far from the axis, or the governor would not be stable.

The problem is not to secure absolute isochronism, but to keep the variation in speed within certain limits chosen with reference to the requirements of the kind of machinery driven by the engine, combined with power and stability. The weighted pendulum, or Porter type, is the simplest

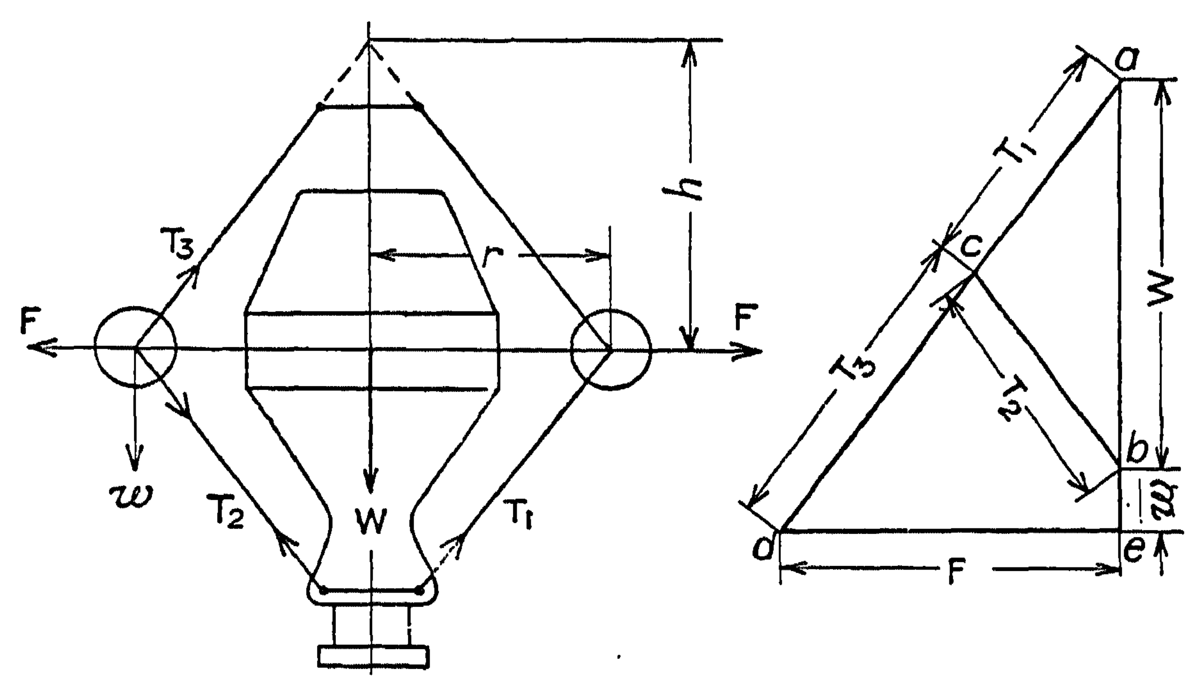


Fig. 22.—Force Diagram for Porter Governor

solution. In its ordinary form a heavy mass is attached to the sliding collar on the spindle, and the weight of the mass is supported by the centrifugal force generated in the balls. If the same range of movement is given to the collar, as in the simple type, obviously there is more stored work available.

It is necessary, therefore, to give to the balls a velocity sufficiently high to enable them to support their own weight and that of the central mass by the centrifugal force generated in them.

Assuming, for both the open- and the crossed-arm type, that the arrange-

ments of links above and below the balls are symmetrical, then the centre weight has double the vertical movement of the balls, and in making calculations each ball is assumed to have a mass equal to the centre weight pulling it down, but not rotating with it, in addition to its own weight. Figs. 20, 21.

Let w lb. be the weight of each ball, and W lb. the weight of central load. The forces acting on the collar are the load W and the tensions T_1 and T_2 on the links. *abc* is the force diagram for these. The forces acting on each ball are the weight of ball w, the tensions T_2 and T_3 in links, and the centrifugal force F. *cbed* is the force diagram for these four forces. Fig. 22.

If h is height and r is radius of cone, then by similar triangles:

$$\frac{W + w}{F} = \frac{h}{r};$$

$$\therefore \frac{W + w}{\frac{w}{g}\,\frac{4\pi^2 n^2 r^2}{r}} = \frac{h}{r};$$

$$\therefore h = \frac{g}{4\pi^2 n^2}\,\frac{W + w}{w}$$

$$= \frac{35{,}200}{n^2} \times \frac{W + w}{w},$$

if h is in inches and n is revolutions per minute; i.e. height of cone for loaded governor = height of cone for unloaded governor $\times \dfrac{W + w}{w}$.

It is obvious then that any increase given to the value of central load W will increase the value of h. The sensitiveness of the governor remains the same, however. Let n and N be the speeds for an unloaded and loaded type respectively for a height of cone h.

Then $\quad h = \dfrac{35{,}200}{n^2}$ (unloaded type),

$$h = \frac{35{,}200}{N^2} \times \frac{W + w}{w} \text{ (loaded type);}$$

$$\therefore \frac{35{,}200}{n^2} = \frac{35{,}200}{N^2} \times \frac{W + w}{w},$$

or $\quad N^2 : n^2 = (W + w) : w.$

Suppose the range of speed for an unloaded governor was from 60 to 70 r.p.m. Then for lowest position of loaded governor, taking W = 40 lb. and $w = 3$ lb.,

$$N^2 : n^2 = (W + w) : w$$

$$N^2 : 3600 = 43 : 3$$

$$N^2 = \frac{3600 \times 43}{3};$$

$$\therefore N = 227 \text{ r.p.m.}$$

For highest position:

$$N^2 : n^2 = (W + w) : w$$
$$N^2 : 4900 = 43 : 3$$
$$N^2 = \frac{4900 \times 43}{3};$$
$$\therefore N = 265 \text{ r.p.m.}$$

Sensitiveness for unloaded type $= \frac{70 - 60}{65} \times 100 = 15{\cdot}4$ per cent,
and for loaded type $= \frac{265 - 227}{246} \times 100 = 15{\cdot}4$ per cent.

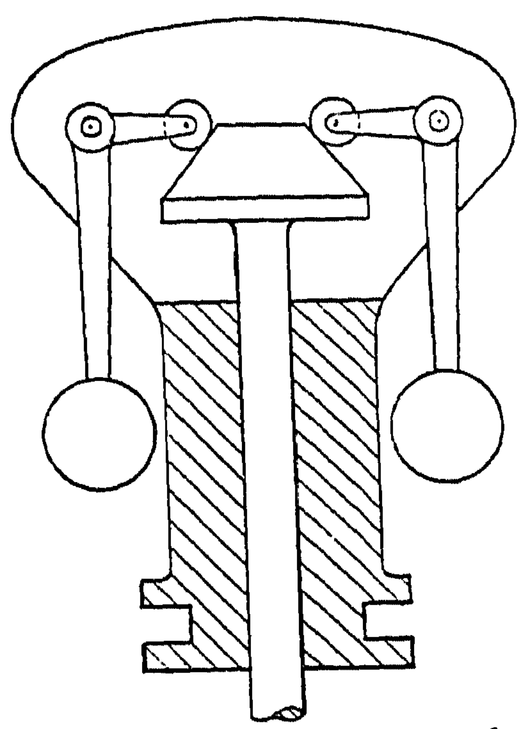


Fig. 23.—Loaded Type of Governor

The loaded governor is therefore not more sensitive than the unloaded type. It is simply more "powerful", and can exert a bigger pull or push. Loading a governor is an artifice for removing the natural range of operation of a high-speed governor where the height and the ratio *change of height/change of speed* is small, to a lower position of the arm where that ratio is greater. The higher speed of rotation gives greater centrifugal force, and this, together with the greater change of height for any chosen percentage of variation, gives more "power".

Sometimes a spring is used, either instead of a control weight or supplementary to it. This arrangement diminishes somewhat the sensitiveness of the governor, as the ratio *change in height/percentage change in speed* is less.

Taking similar governors in the lowest position, the force on the collar, to maintain equilibrium with the centrifugal forces in the balls, must be the same whether springs or weights or both be

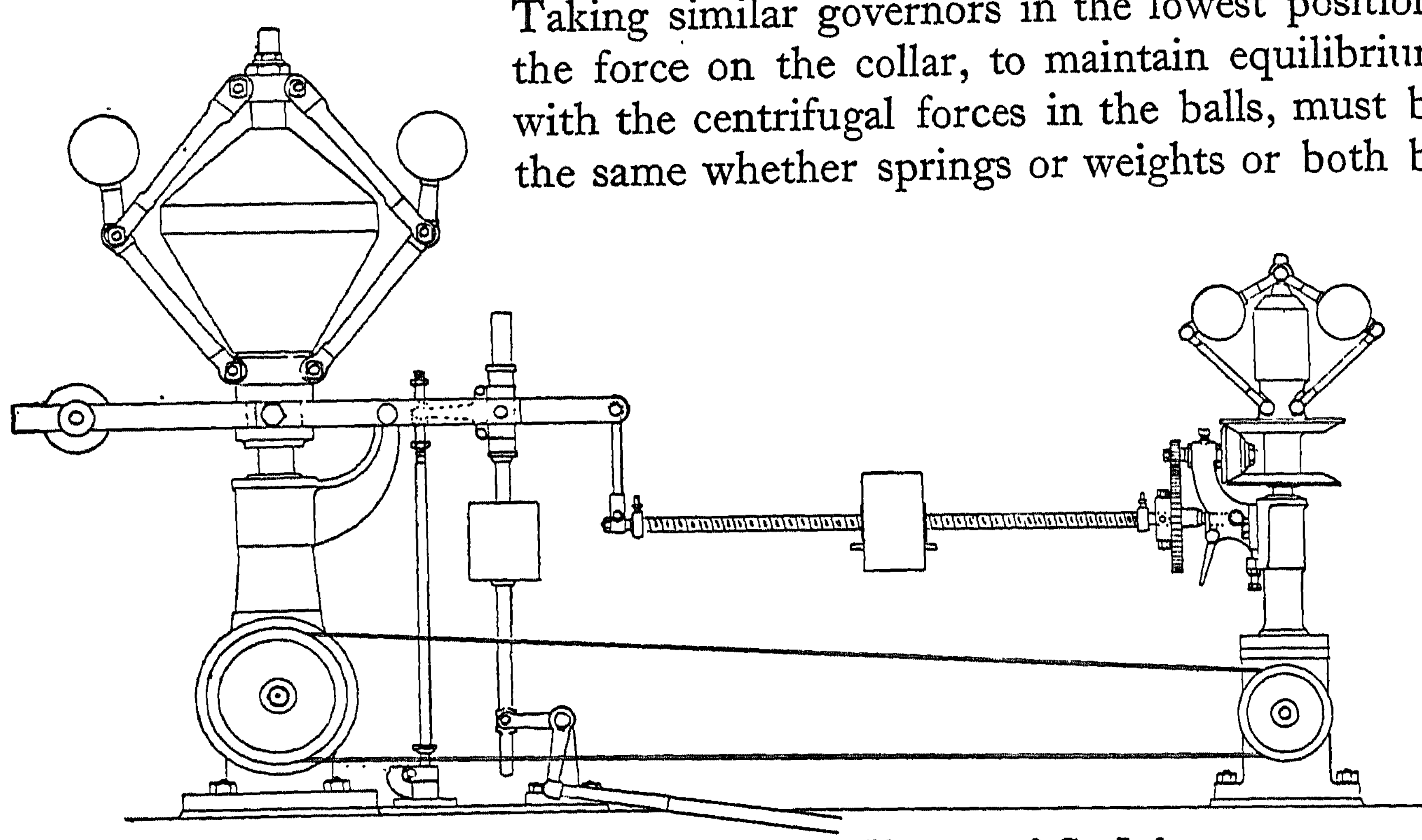


Fig. 24.—Governor Mechanism by Hicks, Hargreaves & Co., Ltd.

used. A spring must have a certain initial compression or tension to give this force, and as the balls move outwards the force exerted by the spring

increases, necessitating a greater alteration in speed for any given movement of the collar. Increasing the weight of the balls would not help, obviously. There are many types of spring governor which may be made as isochronous as desired, but have little power, but we are here concerned with governors intended to be used to actuate valves and gear offering considerable resistance to movement.

There are innumerable designs of loaded governor, some of which are of doubtful merits. One commonly used, fig. 23, has the arms, which are of bell-crank form, pivoted on the central weight, the inner ends being provided with rollers moving along an inclined path. In this case also the path of the balls is approximately parabolic, but the power of such a governor is small, as the weight has little movement and not much work can be stored in it.

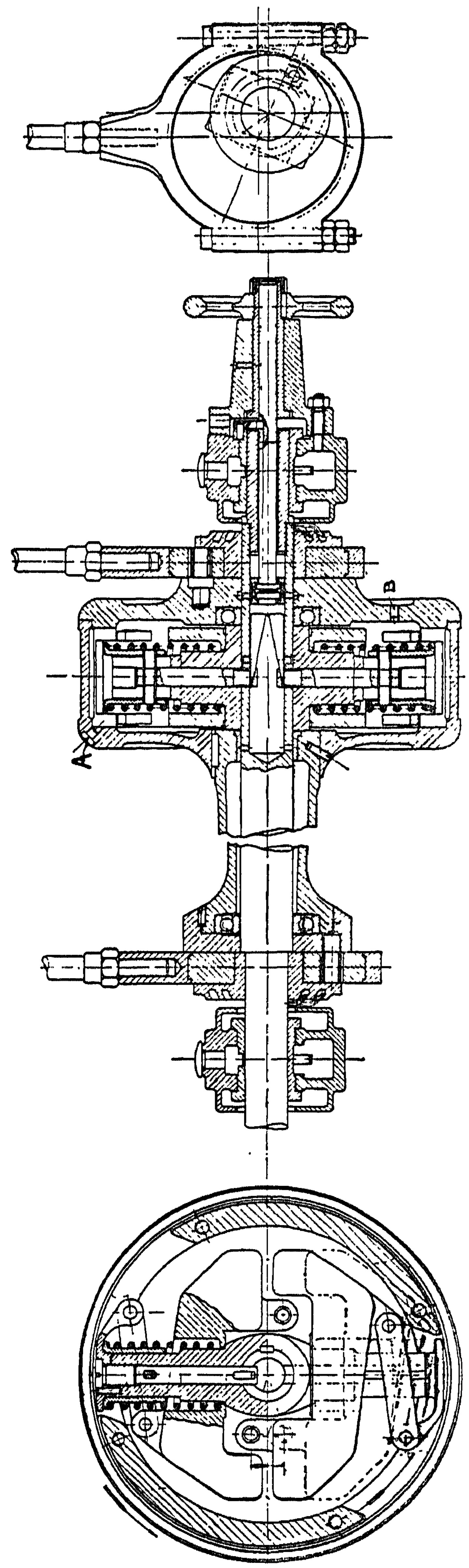


Fig 25.—Robey Crank-shaft Governor

Although isochronism is easily obtained, it is at the expense of stability, and too high a degree of sensitiveness is inadvisable. It is useless to design a governor to work within, or even to approach, the cyclic irregularity of the engine itself or the governor will be continuously on the move.

The phenomenon known as " hunting ", which consists of a rhythmic variation of speed, may occur. The governor moves throughout its range at each change, and steam is alternately fully admitted and completely shut off. Thus racing is followed by undue retardation, but hunting may exist in all degrees. In order to combine stability with a reasonably high sensitiveness, dashpots are often added, in which fluid friction is used to prevent over-hastiness in responding to trivial disturbances, whilst permitting a ready and definite response to even a slight permanent variation of load.

An ingenious method of obtaining practically perfect isochronism combined with stability, a condition useful, for example, in engines driving cotton-spinning machinery, is by employing two governors, both having a fairly high degree of stability. One is large and powerful, and actuates the regulating mechanism direct. The supplementary governor has two

collars on its spindle, between which a friction wheel can revolve. This is mounted on an auxiliary shaft, in such a way that either of the collars can be made to come into contact with it in accord with the sense of the change of speed of the engine. Connected to the shaft, and revolving with it, is a long screw upon which a weight rides, its position on the screw depending upon the action of the supplementary governor. The end of the revolving screw is carried in a bearing, suspended from the lever of the main governor, and the varying position of the weight along the screw modifies the effect of the centre weight of the governor upon the balls.

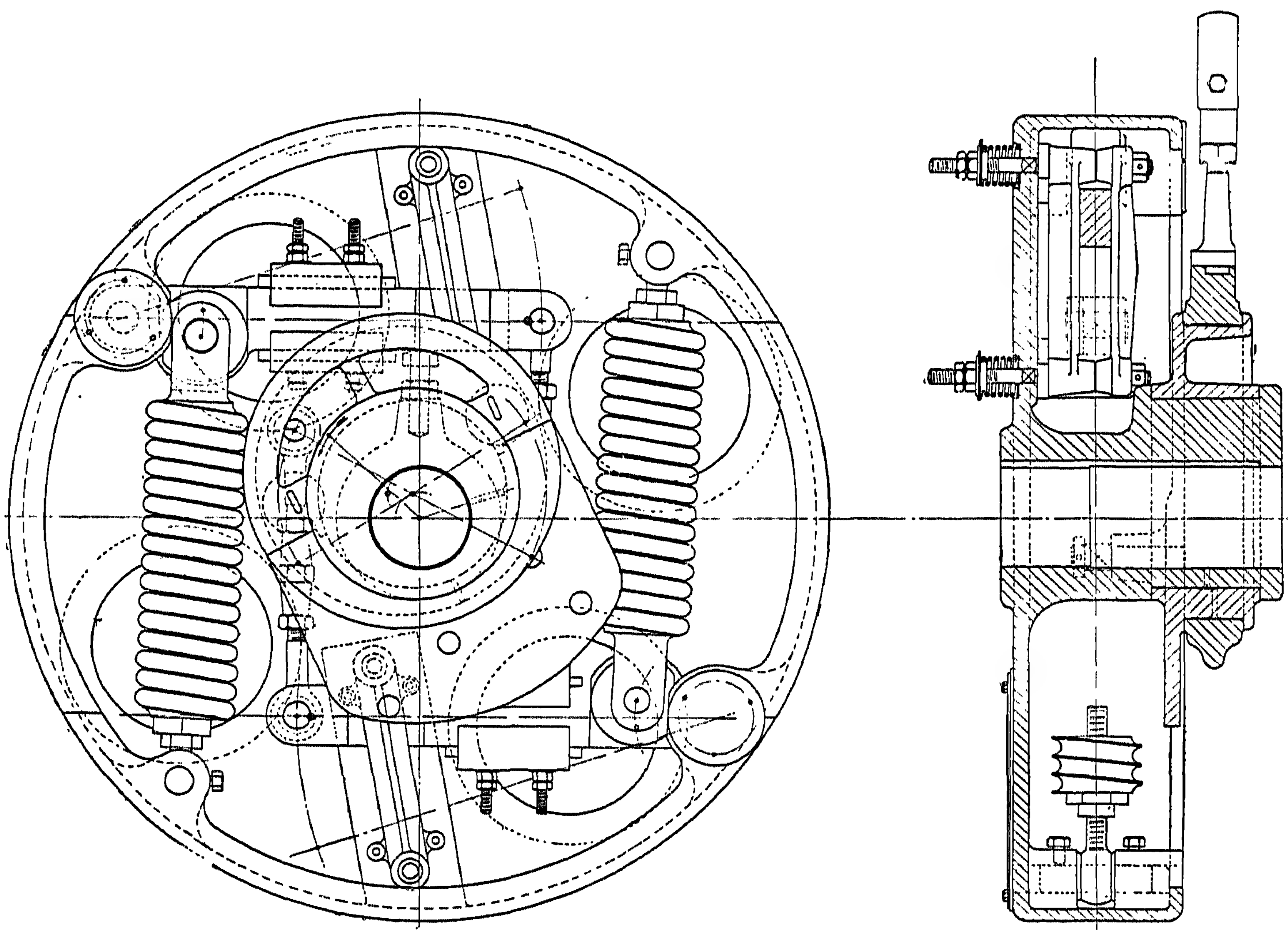


Fig. 26.—Crank-shaft Governor

Extremely accurate regulation is obtained by this means for gradually varying loads, but for sudden variations the sensitiveness is that of the main governor alone. The arrangement is shown in fig. 24.

With the advent of the Uniflow type of engine in which the speed of revolution is not high, crank-shaft governors have again come into favour for regulating the cut-off with drop valves, more especially when the so-called positive type of valve gear is used.

The general principle upon which they are designed is that the angle of advance of the eccentric is modified by the movement of weights, the centrifugal force of which is controlled by springs. An example of this type of governor, made by Messrs. Robey & Co., is shown in fig. 25.

This governor is fixed upon the lay-shaft from which the valve gear is driven.

Another example, by John Musgrave & Co., Ltd., of Bolton, is shown

in fig. 26. This governor is mounted upon the crank-shaft on the side remote from the driving end, and the eccentric drives the valve gear through a rocking shaft fixed upon the frame.

Diagrams such as fig. 43 (p. 128) may be used for investigating the properties of any kind of governor, whether working in a horizontal or a vertical position, and whether controlled by springs or by gravity or by a combination of both.

CHAPTER II

Marine Engines

Introduction.—Although the reciprocating marine engine has been submitted to very keen competition from the steam turbine, and more latterly from the internal-combustion engine, there are many engineers who still prefer the reciprocating steam-engine for medium and low-powered vessels on the grounds of reliability and general convenience in working. Even on the score of total running costs, including coal consumption, which is the final test applied by shipowners, a well-designed quadruple-expansion job can compete not unfavourably with geared turbines in sister ships when all-round expenses are taken into account, whilst capital cost is considerably less.

The modern marine engine has become very much standardized, and the various makes differ from each other principally in minor points.

The usual type of engine for a modern-sized cargo-steamer is of the triple-expansion vertical design, the quadruple-expansion type being used in intermediate boats carrying both passengers and cargo.

The cylinders are invariably placed above the crank-shaft and carried on columns supported by the bedplate which carries the crank-shaft. The air pumps, often of the Edwards type, are driven from one of the main cross-heads, and the condenser is now always a separate cylindrical structure usually of steel plate and supported from the back columns, but there are various special makes of condenser in the market.

The Michell thrust block is largely displacing the horse-shoe collar type which has been so long in use.

Steam pressures have gradually increased, and are 160 to 200 lb. per square inch for triple-expansion, and 205 to 220 for quadruple-expansion engines.

Superheat has come into extensive use, giving considerably improved economy.

The ratio of the low-pressure to high-pressure cylinder is from 6·5 to 8 for triple-expansion engines and from 8 to 9·5 for quadruple-expansion engines with the intermediate cylinders geometrically proportionate. The mean pressures in pounds per square inch referred to low-pressure cylinders

are from 30 to 40 for triples, and from 33 to 38 for quadruples, but these greatly depend upon the class of vessel. The revolutions per minute are from 75 to 85 for mail boats and 65 to 90 for cargo boats. The piston speeds in feet per minute are from 750 to 950 or 1000 for mail boats with balanced engines, and from 500 to 750 for cargo boats.

Cylinders.—The cylinders, except for the smallest sizes, are always cast separately. In high-class work the cylinders are each provided with a liner of specially hard, close-grained cast iron. The high-pressure cylinder has invariably a piston valve, and the intermediate-pressure cylinder often has some form of balanced slide valve. Low-pressure cylinders have usually a double-ported slide valve, but in some cases piston valves are fitted to all cylinders.

The rules for the thickness of the cylinder walls and of the liners are usually based upon practice, and vary greatly. The question of strength to resist the bursting effect of the steam pressure is obviously the first consideration, especially for the high-pressure cylinder, but allowance must be made for wear and the probable necessity for reboring, and also for the possibility of the casting being thinner in places than intended, due to the core being out of centre. If the low-pressure cylinders were designed only to resist the steam pressure, the metal would come out very thin. Structural stiffness is necessary, and practical considerations in making the casting must be regarded, so that it is usual first to fix the thickness of the high-pressure cylinder and then to make that of the others the same throughout.

Good cast iron, such as is used for cylinders, should bear a stress of 2200 to 2400 lb. per square inch. The thickness of the high-pressure cylinder may be, therefore, boiler pressure $\times \dfrac{\text{diameter of cylinder}}{4600} + \frac{1}{4}$.

For the liner the same rule may be used, but the constant added may be $\frac{1}{4}$ to $\frac{3}{8}$ in. to allow for reboring. If no liner be used, the thickness of the cylinder barrel may be about the same as that for a liner.

A rule sometimes used is to allow a stress of 1600 lb. per square inch for the high-pressure cylinder, 1000 lb. per square inch for the intermediate-pressure, and 550 to 600 lb. per square inch for the low-pressure. Full boiler pressure is assumed on the high-pressure, 70 lb. per square inch on the intermediate-pressure, and 25 lb. per square inch on the low-pressure cylinders. These figures allow for reboring.

The valve chests are usually cast with the cylinders. When piston valves are used they are provided with liners of hard, close-grained metal for the piston valves. When slide valves are used in the low-pressure cylinder, a face of similar metal is used. The ports are kept as short as possible with the object of decreasing the clearance volume.

The speed of the steam through the ports is 5000 to 5500 ft. per minute for the high-pressure cylinder, 6500 to 7000 ft. per minute for the intermediate, and 8000 to 8500 ft. per minute for the low-pressure. The speed through the holes in the valve liners is necessarily higher. The valve diameter and the valve travel should be such that the speed of the entering steam should

not be more than 6000 ft. per minute in the case of high-pressure cylinders, and 9000 ft. per minute for the intermediate-pressure cylinders. The low-pressure cylinder ports and port opening often present a difficulty, and the speed through the latter may need to be as high as 14,500 ft. per minute, 11,000 to 12,000 ft. per minute being good figures.

Steam jackets are not now usually fitted, even though liners may be provided for the high-pressure and low-pressure cylinders. Experience has shown that the loss by condensation in the jackets is about equal to the

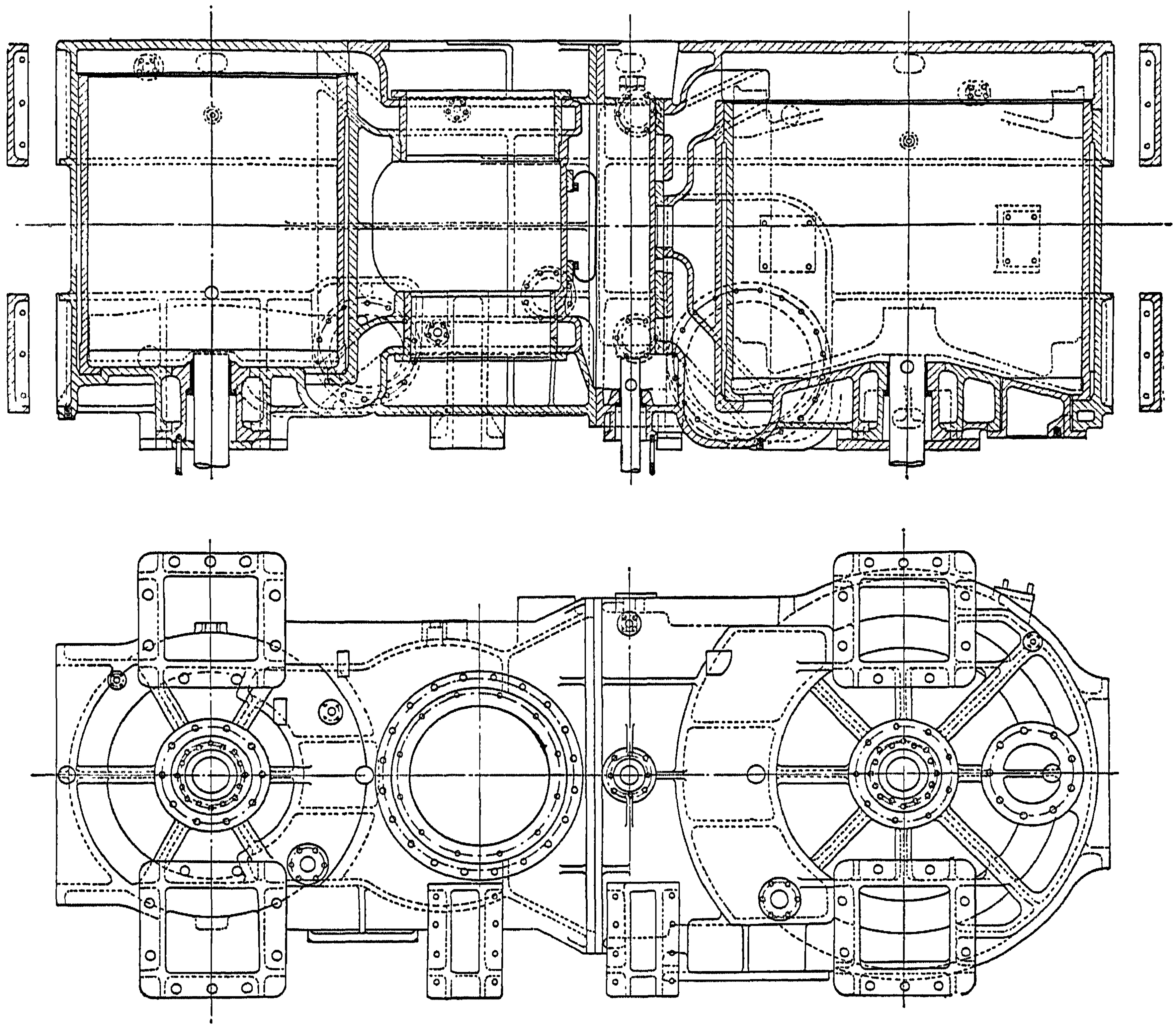


Fig. 27.—Marine-engine Cylinders

saving effected in the cylinders by the presence of steam in the jackets. Warming valves for use before starting the engine are usually fitted and steam allowed to enter the space between the liner and the cylinder barrel. This space should, of course, have arrangements for its being effectively drained to prevent accumulation of water. A waterlogged jacket would do positive harm. There are many ways of securing the liners in the cylinder bodies; a typical method is shown in fig. 27. The bottom cylinder covers and stuffing-boxes are cast with the cylinders, and, of course, follow the shape of the under side of the piston, and are well ribbed on the under side. The feet which rest upon the columns are also cast with the cylinders, and are tied to the barrel by ribs carried well up. Sometimes the feet are of hollow box-section, giving a neater appearance. The column-bolts are

from four to six in number, and as they have to sustain heavy shocks, owing to the occasional pressure of water in the cylinders, the stress in the bolts at the bottom of the thread is kept low, say from 3000 to 3500 lb. per square inch. Sometimes all these bolts are turned and fitted into reamered holes, but some makers fit only two turned bolts, those on opposite diagonal corners. The cylinder foot may have a thickness to $1\frac{1}{4}$ to $1\frac{1}{2}$ times the diameter of column-bolt.

The cylinder covers are of heavy ribbed- or box-section, special attention being given to the low-pressure cover to avoid light sections which may cause unsightly "breathing" as the steam is admitted into the cylinders. The shape of the covers follows that of the top side of the piston.

If a liner is fitted to the cylinder when hot, the under side of that part of the cover which projects into the cylinder is made just to clear the top of the liner, allowing, of course, for the thickness of jointing material, thus helping to keep the liner in position.

The thickness of metal in the cover may be about 0·6 to 0·8 times the thickness of the cylinder, the usual conical shape of the cover giving ample strength. The cover flange may have a thickness of $1\frac{1}{4}$ times the diameter of studs. The size of the studs is usually determined for the high-pressure cylinder cover and made the same for all the covers, including those for the valve chests, enabling the same spanner to be used throughout. Full boiler pressure is assumed acting upon the diameter of the stud circle, and a diameter of stud chosen such that their number will give a pitch of from 3 to $3\frac{1}{2}$ times the diameter of stud. Studs having a diameter less than $\frac{3}{4}$ in. in the body should not be used, and the stress at the bottom of the thread should not be more than 4000 lb. per square inch for the smaller diameters, or more than 5500 lb. for the larger. The pitch of the studs in the intermediate-pressure and low-pressure cylinders may be from $4\frac{1}{2}$ to 5 diameters and 6 to 7 diameters respectively. The internal projection of the cover is well cut away opposite the steam port, to allow an easy flow for the steam. The larger cylinder covers often have an inspection door or manhole in the centre of about 15 or 16-in. diameter, to permit access and inspection. This door is, of course, placed off the centre in the bottom covers. Bosses for relief valves are provided, the valves having a diameter of $\frac{1}{12}$ to $\frac{1}{8}$ the diameter of the cylinder. There are also bosses for drain and indicator cocks.

When designing the cylinders, provision for attaching the lagging sheets should not be forgotten.

The clearance between the piston and the covers is $\frac{3}{16}$ to $\frac{1}{4}$ in. at the top and $\frac{1}{4}$ to $\frac{3}{8}$ in. at the bottom for cylinders of 16 to 24 in. diameter, $\frac{3}{8}$ to $\frac{1}{2}$ in. at the top and $\frac{1}{2}$ to $\frac{9}{16}$ in. at the bottom for cylinders of 40 to 60 in. diameter, and $\frac{1}{2}$ to $\frac{9}{16}$ in. at the top and $\frac{5}{8}$ to $\frac{3}{4}$ in. at the bottom for cylinders of 80 to 100 in. diameter.

The steam ports have one or more internal strengthening ribs connecting the top flange with the body of the cylinder.

Valves.—The usual practice for triple-expansion engines is to equip

the H.P. and I.P. cylinders with piston-valves, and to fit slide-valves to L.P. cylinders, except for fast-running engines, which often have piston-valves fitted throughout, as the power required to operate the slide-valves is considerable. When the slide-valves and gear are heavy, a balance cylinder is used to take up the weight, and in very large engines what are called "assistance" cylinders are used. They actually help to drive the valve and relieve the eccentrics and gear of much work.

There is not much variation in the design of piston-valves. Liners are invariably used. The steam may be admitted between the working ends of the valve or at the extremities. In the first case, which is the usual arrangement, the valves are said to have "inside" admission and the eccentrics are mounted on the shaft at an angle of 180° to the normal arrangement for a slide-valve. In the other case the valves have

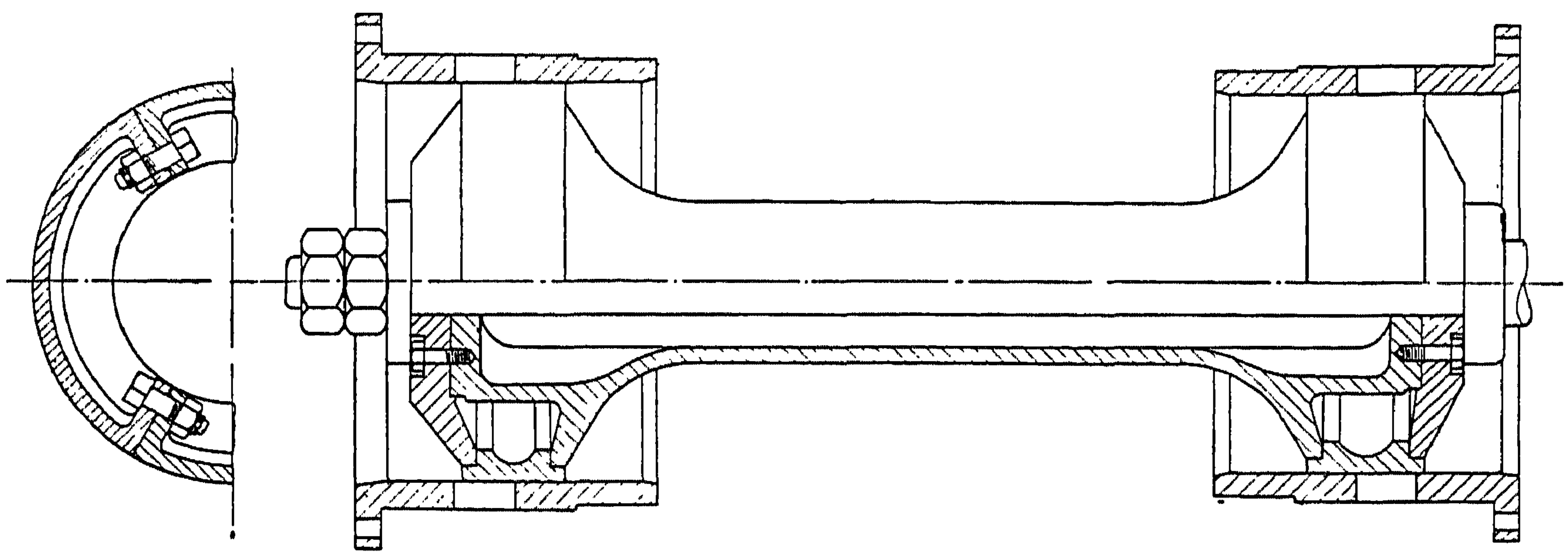


Fig. 28.—Piston Valve

"outside" admission. "Inside" admission has the advantage that the valve-rod packing is subjected to exhaust pressure only. The valves are sometimes made hollow to allow the exhaust steam from one end to pass through them to the exhaust outlet. But from a thermal point of view this is a bad arrangement because of the exchange of heat between the exhaust steam and the inlet steam.

Valving packings are of many types. Plain floating rings are often used, but one of several well-known proprietary types which have proved satisfactory in service is often specified. Fig. 28 shows an example of piston-valve design of Messrs. Scott & Co.

Columns.—These are usually of cast iron, of plain shape tapering from the smallest section at the top, and of rectangular box-section. The stress at the smallest section is kept low, from 500 to 600 lb. per square inch as the load alternates from tension to compression, and water in the cylinder may cause severe shocks. Two columns are used for each cylinder, one on each side of the crank-shaft. In some cases the front columns are of forged steel. These have to sustain an alternating load, and are therefore in much the same condition as the piston-rod, but are longer. On the other hand, they have to sustain only half the piston load, and are well secured at the ends, so that their diameter may be based upon a stress of

about 1000 lb. per square inch due to the piston load. The stress allowed upon the column-bolts is 3000 to 3500 lb. per square inch for the connection to the cylinders, and from 2500 to 3000 lb. per square inch at the foot for the connection to the bedplate.

Bedplate.—The bedplate is usually made of cast iron built up of several sections which are bolted together with heavy flanges. It contains

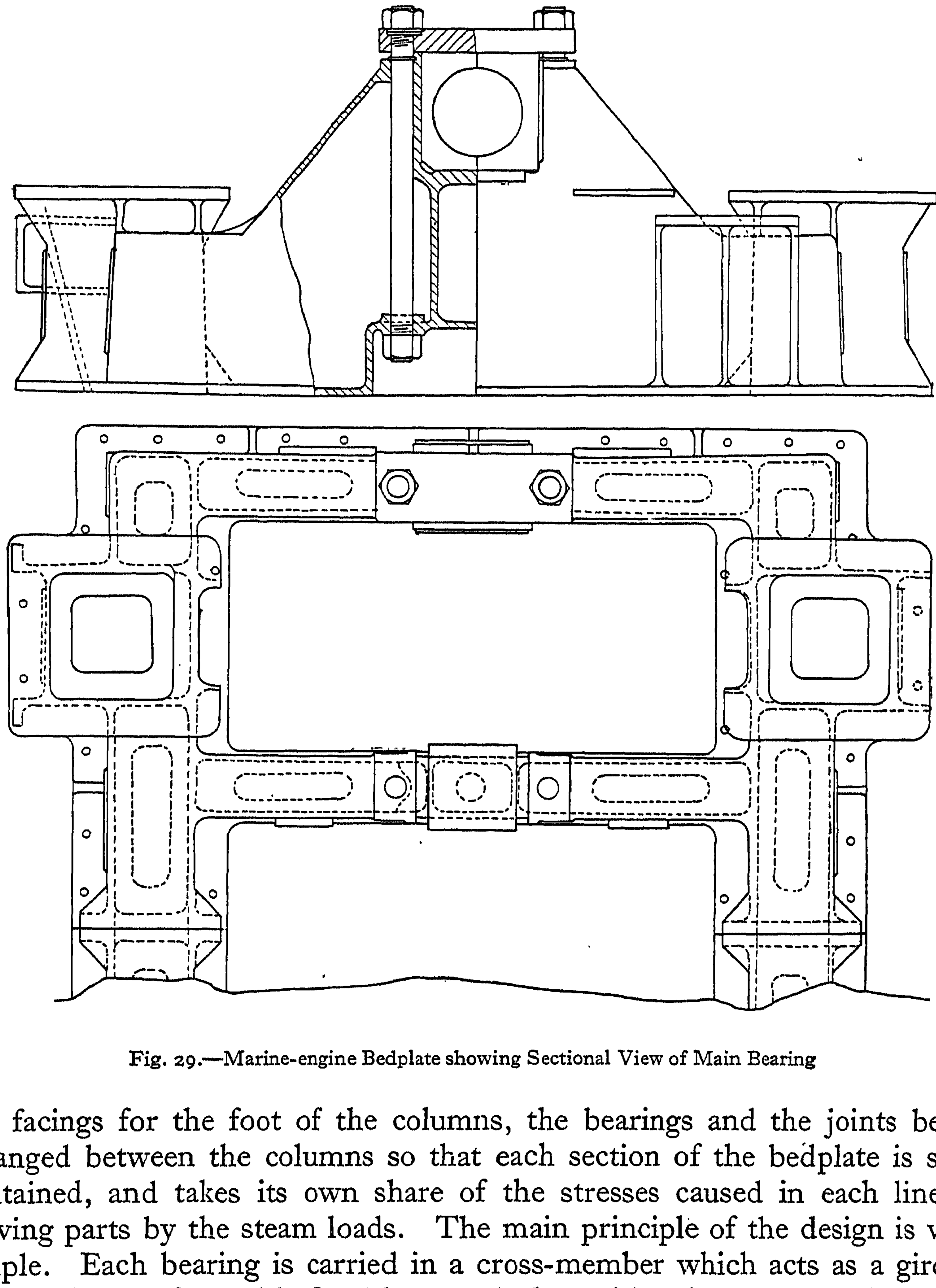


Fig. 29.—Marine-engine Bedplate showing Sectional View of Main Bearing

the facings for the foot of the columns, the bearings and the joints being arranged between the columns so that each section of the bedplate is self-contained, and takes its own share of the stresses caused in each line of moving parts by the steam loads. The main principle of the design is very simple. Each bearing is carried in a cross-member which acts as a girder. The thickness of metal is fixed by practical considerations, and as the depth at the centre is sufficiently great to give a reasonable clearance between the working parts, when at the bottom of the stroke, and the foundation of the engine, the bedplate is exceedingly rigid and strong. The main bearings

ually of cast iron, of plain square form lined with white metal; the ess top and bottom without the white metal may be $\frac{D}{8} + \frac{1}{4}$ where D in diameter of the crank-shaft, the thickness at the sides being about that figure. The cap bolts extend right down to the bottom of edplate, a recess being formed to accommodate and to give access nuts at the bottom. The whole depth of the section of the bed- at the centre is thus available to resist the stresses set up during the oke of the engine, which would not be so if studs or T-headed bolts n the metal at the top near the bearing were used. The stress on lts at the bottom of the thread is on account of their length kept low, not exceeding 3500 to 4000 lb. per square inch. Collar-nuts fitting cesses in the cap secured by set screws are used at the top end, and crew through one of the flats of the nut is used at the bottom end uring the nuts. The cap is always made of mild steel, and its thick- nay be $\frac{1}{8}$ to $\frac{1}{4}$ in. greater than the er of the bolts. A sectional view main bearing, such as described is shown in fig. 29.

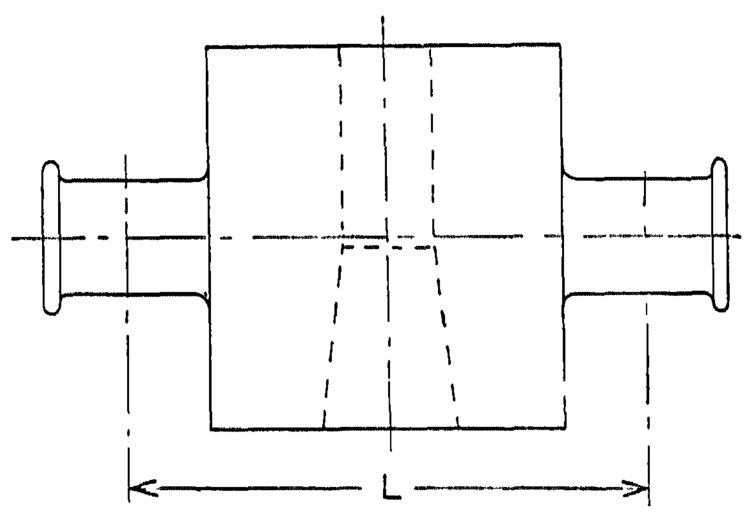


Fig. 30.—Crosshead Block

ossheads and Guides.—Cross- may be divided into two classes, for which the guides are supported back column only, and those for the ahead guide is fixed on the olumn and the astern guide is fixed front column, the former type having a single slipper and the latter ppers.

ommon type of crosshead takes the form of a cubical forging having gudgeon pins solid with it. The piston-rod is attached to the cross- n exactly the same way as to the pistons, that is, by a taper part fol- by a parallel part with a screw thread and nut, the diameters of these s being exactly the same as at the piston end of the rod. The er of the gudgeon pins may be about the same as the diameter of the rod, or even 25 per cent more, and the length such that the total from 800 to 1000 lb. per square inch upon the bearing surface. The usually comes out about equal to the diameter. These proportions give security against bending, but the block itself, fig. 30, is subjected to g stresses through the mid-section at right angles to the paper, the g moment being $\frac{WL}{4}$ when W lb. is the load upon the high-pressure and L in. is the distance between the centres of the gudgeon pins. s of 6000 lb. per square inch may be allowed at the outer layers of tion, as the load is in alternate directions.

slippers, usually of cast iron, are attached to the crosshead block screws or "tap bolts", four in number, and are about one-fourth of meter of the screwed end of the piston-rod. Lips or projections on

the slippers between which the crosshead block fits are provided to take the shearing caused by the inertia and friction of the slippers. The working face of the slipper usually consists of white metal, cast in grooves on the metal, often in the form of strips, the surface being such as to give a pressure of 50 to 70 lb. per square inch taken when the crank is at right angles to the line of stroke.

When there is only a single guide, the pressure on the guide strips for astern working should not exceed 60 to 80 lb. per square inch. With this type of guide the strips are subject to a force tending to tear them off the supporting bolts when the ship is running astern, and they should be designed to resist the bending action set up. They are usually of cast iron,

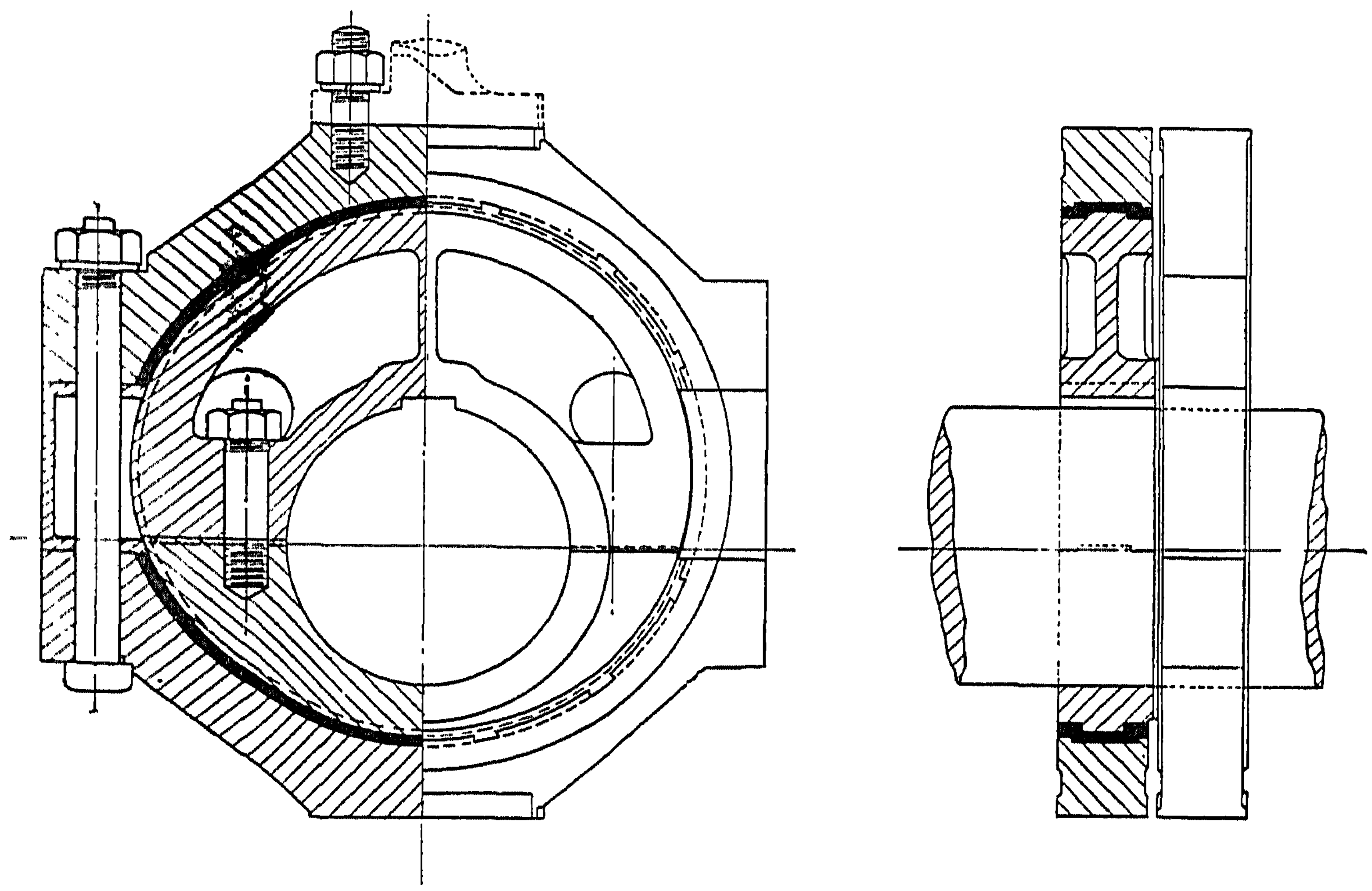


Fig. 31.—Eccentric Pulley and Straps

and the stress should be kept low, say 1500 lb. per square inch, especially as there is a sharp internal corner. The bolts should have a stress of 3500 to 4000 lb. per square inch at bottom of thread. They are spaced about 6 to 7 diameters apart, and all of them may be taken as acting together to resist the lifting action of the connecting-rod.

Eccentrics and Valve Gear.—The eccentric pulley is of course split, both parts being often of cast iron, fig. 31. The thickness at the centre of the smaller part may be $\frac{1}{6}d + \frac{1}{4}$ in. (d in. being the diameter of shaft). The two parts are fixed on the shaft by studs which are screwed into the smaller part, a single nut often being used at the other end. The diameter of these studs may be based on the loads used for designing the valve gear. The two parts are held in place laterally by a groove-and-feather joint, the feather usually being formed on the smaller part. In small engines there may not be room for nuts on the ends of the studs, and it would then be necessary to use cotters. The eccentric straps are also made of cast iron, and lined with white metal in the usual way, and are usually made of plain rectangular section with corners rounded. The breadth of the eccentric pulley

and straps should be such that the maximum pressure does not exceed 100 to 120 lb. per square inch with a load on the eccentric rod, which is taken as the area of the whole face of the low-pressure slide valve × 30 lb. per square inch × 0·2. The factor 0·2 is the value of the coefficient of friction. The bending stress, due to the above-mentioned load, should be calculated for one or two sections of the strap, and its radial thickness should be such that the stresses are kept low to prevent the strap closing in upon the pulley on the downward travel of the valve, which would cause the eccentric to run hot. The foot of the eccentric rod is secured to the top half strap by studs, the diameter at the bottom of the thread being based upon the steam load on the low-pressure valve, allowing a stress of 4500 to 5000 lb. per square inch. The studs holding the two parts of the pulley together may

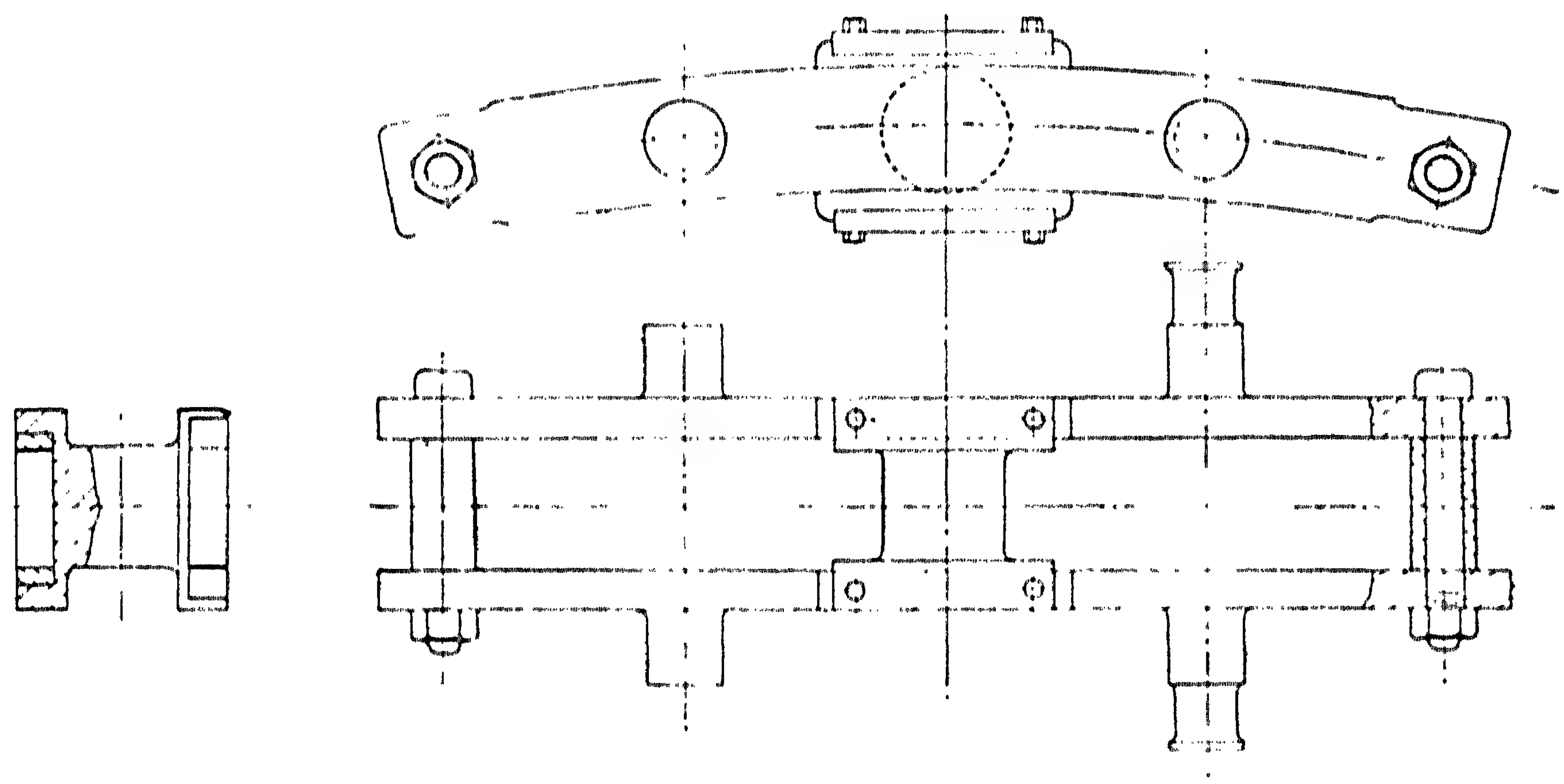


Fig. 32.—Bar Expansion-link

be a little larger in diameter to allow for occasional rough usage, and the bolts which hold the two halves of the straps together may on account of their length be also slightly larger in diameter.

The bar expansion link, fig. 32, is the type almost invariably adopted. Two bars or quadrants are used, each having two gudgeons forged on, one for the ahead eccentric rod, and the other for the astern rod. The ends of the eccentric rod are, of course, forked and provided with bearings to take the corresponding gudgeons on each quadrant. The distance apart of the two gudgeons is usually 6 times the throw or radius of eccentric. Between the quadrants is placed the saddle block through which the quadrant bars slide, the brass liners for the working faces being fixed in the jaws of the saddle blocks. The length of the working face is usually about equal to half the distance between the gudgeon centre lines. The pressure on the face should be from 300 to 350 lb. per square inch In computing the stresses on the quadrants due to the force required to drive the valves, the bars are treated as beams loaded at the middle and supported at the ends. The bar is bent alternately in both directions, and therefore, in order to secure stiffness also, the stress allowed is not more

than 5000 lb. per square inch. The thickness of the links is about $\frac{1}{3}$ of their depth. The diameter of the eccentric rod gudgeons may be about $\frac{3}{4}$ of the depth of the quadrant bar, and the diameter of the drag link pins may be 0·7 of this diameter.

The stress on the valve-rod screw may be 5000 lb. per square inch and 2800 to 3000 lb. per square inch in the body. The diameter of the part passing through the stuffing-box is sometimes half the diameter of the piston-rod + $\frac{1}{2}$ in. The stress on the valve-gear bolts may be 4000 to 4500 lb. per square inch. The bearing pressure on the valve-spindle eye may be from 600 to 700 lb. per square inch, and on the eccentric-rod pins from 700 to 850 lb. per square inch. All the stresses and bearing pressures are computed on the load on the low-pressure valve given above.

Pistons.—Pistons are usually made of cast iron of heavy box-section, but low-pressure pistons and those for the second medium-pressure cylinders

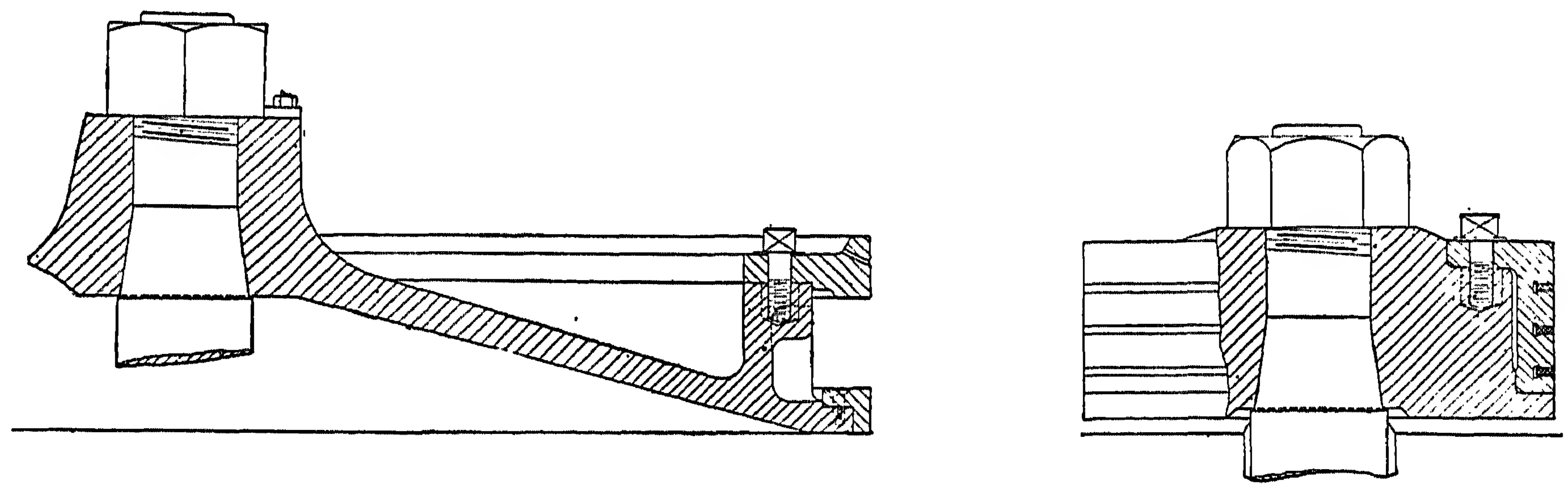


Fig. 33.—Typical Piston Packing Rings and Methods of Fastening in Marine Practice

of quadruple engines may be made of cast steel, or sometimes of forged steel. The high-pressure piston for small cylinders is often solid. Various types of packing rings are used, but the simplest and on the whole the most satisfactory is a plain Ramsbottom ring of narrow face and rectangular section, which will not wear the cylinder. The rings are turned to a diameter of about $\frac{1}{64}$ in. larger than the bore of the cylinder for each 1 in. diameter. They are then cut and compressed until the joint is open about $\frac{1}{32}$ in., and are then turned to fit the cylinder bore. The axial thickness of the rings may be $\frac{3}{8}$ in. for the high-pressure cylinder to 1 in. for the low-pressure cylinder, three rings being used for each piston. The radial depth of the rings may be from 1 to $1\frac{3}{8}$ in. The rings are inserted in a separate carrier which can be withdrawn for examination independently of the piston. This is secured to the piston body by steel screws with square heads screwing into gun-metal nuts, which are themselves screwed into the piston body. An example of this construction is shown in fig. 33. Brass washers are placed underneath the heads of the screws. In many cases the piston rings are made by firms who manufacture them as specialities. The restrained type of ring is often used for high-pressure cylinders. They are always fixed in a carrier ring.

It is not easy to apply calculations for the strength of pistons to such a structure as a hollow box-section, and the only guide is experience.

With regard to the cast-steel low-pressure piston and the second intermediate pressure piston, the thickness in inches of the metal near the boss may be $\frac{D}{50} + 0{\cdot}8$ and $\frac{D}{34} + 0{\cdot}7$ respectively, the thickness near the rim being $0{\cdot}7$ of the thickness near the boss. D is the diameter of the cylinder.

Connecting-rod.—The connecting-rods are from four to six cranks long, 5 : 1 being a very suitable ratio. The middle of the body of the rod should have a diameter not less than that of the piston-rod, and the diameter of the top end near the fork should be such as to give a stress of not more than 5000 lb. per square inch, as the stress is alternating in direction. The rod usually has a straight taper to the foot. These dimensions are sufficient to make the rod stable against the combined effects of the axial crippling load and the bending effect of the inertia of the rod as it swings. The gudgeon bearings are of gun-metal, and when large are cored out at the back for lightness. The bolts have a stress of 4500 to 5000 lb. per square inch at the bottom of the thread, and the caps may be $\frac{1}{8}$ to $\frac{1}{4}$ in. thicker than the diameter of the bolts, according to the size of the work. This will usually give the stiffness required and a satisfactorily low stress which should not, however, exceed 8000 to 9000 lb. per square inch, taking the cap as a beam with a uniform distributed load. The nuts are secured by locking screws, the points of which engage in a groove cut in a cylindrical portion turned on the nut, which projects into a suitable recess in the cap. A liner is usually fitted between each pair of brasses at the joint to permit of adjustment. Sometimes this liner, instead of being solid, is composed of one thick piece with a number of other pieces of varying thicknesses down to $\frac{1}{64}$ in. in thickness. The fork at the top end of the rod is usually Y-shaped with straight sides to suit the shape of the crosshead. The metal forming the sides is subjected to bending stresses in both directions, and the stress should be calculated at one or two sections, taking the bending moment as half the load on the piston multiplied by the distance between the centre of one of the bearings and the neutral axis of the section.

As the stress is of an alternating kind it should be kept low, say 5000 to 6000 lb. per square inch. This also allows for the tensile and compressive stresses. If these are taken into account the stress may be 9000 lb. per square inch. In order to prevent the possibility of the sides of the Y spreading under stress some makers form a collar upon the end of the gudgeons. This certainly is safer. The bottom end of the rod is designed after the diameter and length of the crank pin has been fixed.

The bottom end bearings are often made of steel castings lined with white metal. The bending moment upon the cap in the centre may be taken as $= \frac{PL}{8}$, where P lb. is the piston load and L in. is the distance between the bolt centres. A stress of 7000 to 8000 lb. per square inch may be allowed. A distance piece composed of several liners of brass of assorted thicknesses separates the two halves. The width of the connecting-

rod foot is about 70 per cent of the width of the bearing, and its thickness $\frac{1}{8}$ to $\frac{3}{8}$ in. greater than the diameter of the bolts. The stress on the bolts at the bottom of the thread may be 4500 to 5000 lb. per square inch.

Piston-rods.—These are made of mild steel. The stress in the body of the rod should be 2800 to 3000 lb. per square inch, and the stress at the bottom of the thread for the piston nut should be 5000 to 5500 lb. per square inch. The part which enters the piston should be partly parallel and partly taper, the parallel part having the same diameter as the top of the threads of the screwed part, which usually has four threads per inch. The taper is usually one in four on the diameter, and the larger end of the taper is a little less than the diameter of the body of the rod, leaving a shoulder at that point. The shoulder does not, however, butt against the under side of the piston, a slight clearance of about $\frac{1}{32}$ to $\frac{1}{16}$ in. being left between these two parts. The attachment to the crosshead is frequently precisely similar in design to that at the piston end.

Thrust Block.—Recently a new type of thrust bearing (illustrated in Vol. IV, p. 9), invented by Mr. Anthony G. M. Michell, has come into successful use. There is only one collar, and the length and weight of the whole fitting is much reduced. Its design depends upon the application of principles first laid down by the late Professor Osborne Reynolds as a result of his investigations into the theory of lubrication.

Briefly put, it means that considerable loads per unit of area can be sustained if the two rubbing surfaces are not parallel, but have a slight inclination towards each other in the direction of motion, so that the film of oil between the surfaces is wedge-shaped, the thicker end being at the entrance edge. The two surfaces together form a kind of pump which continually renews the film of oil between them, and by maintaining the pressure of the oil prevents, even under very high loads, metallic contact. Experience showed that one of these surfaces must be free to take up the exact inclination required by the varying conditions of load, surface speed, and viscosity of oil. With these conditions the coefficient of friction is reduced to about 0·0018, or $\frac{1}{20}$ of the value that would exist under similar conditions of loading if the surfaces were maintained parallel to each other.

Crank-shafts.—In practice these shafts are never designed *ab initio*. For many years certain rules laid down by the Board of Trade, and such societies as Lloyd's Register, the British Corporation for Survey and Register of Shipping, and the Bureau Véritas International Register of Shipping, were worked to by designers.

Recently a committee was formed, entitled British Marine Engineering Design and Construction Committee, composed of gentlemen interested in the design and manufacture of marine engines, and of representatives of the societies above mentioned. The object of this committee was to standardize, as far as possible, design and construction, and new rules governing the sizes of shafts, having been agreed upon, were issued.

So far as reciprocating engines are concerned, the rules are as follows:

SHAFTING OF RECIPROCATING STEAM-ENGINES

Reciprocating steam-engine installations shall have shafts the diameters of which shall be not less than as given by the following rules, where:

D is the diameter in inches of the low-pressure cylinder, or the equivalent diameter where two or more low-pressure cylinders are used.
S is the stroke of piston in inches.
P is the working pressure in the boiler in pounds per square inch.
r is the ratio of the swept volume of the low-pressure cylinder or cylinders to that of the high-pressure cylinder. Then

1. Diameter in inches of the intermediate shafts

$$= \sqrt[3]{\frac{D^2 \times S \times P}{f(r + 2)}}.$$

The following are the various values of f:

General Description of Engine.	Service of Ship. No. 1.	No. 2.
(a) 2 Cranks at 90°, Cylinders compound, triple, or quadruple	1,800	2,000
(b) 2 Cranks at 180°, Cylinders compound, triple, or quadruple	1,260	1,400
(c) 3 Cranks at 120°, Cylinders compound, triple, or quadruple	2,000	2,250
(d) 4 Cranks, balanced, Cylinders compound, triple, or quadruple	2,000	2,250
(e) 4 Cranks at 90°, Cylinders quadruple ..	1,900	2,150

No. 1 is for ocean-going and cross-Channel ships such as would have a No. 1 or a No. 2 certificate of the Board of Trade if carrying passengers.

No. 2 is for ships coasting or working on estuaries, rivers, lochs, and lakes where the water is habitually smooth, and which would have the No. 3, No. 4, or No. 5 certificate of the Board of Trade if carrying passengers.

2. Crank-shafts for the various engines shall have a diameter not less than 1·05 times that required for the intermediate shafts.

3. The diameter of thrust-shafts transmitting torque shall not be less in way of the collars than 1·05 times that required for the intermediate shafts. This may be tapered down outside the collars to the size required for the intermediate shafts.

4. The tube-shaft diameter shall be in no part less than 1·05 times that required for the intermediate shaft, and any part of it in the tube exposed or likely to be exposed to sea water shall be not less than 1·075 times that required for the intermediate shafts.

TAIL SHAFTS

Tail shafts within the stern bush, whether in the stern tube or in the bracket, shall be not less in diameter than given by the following:

$$\text{Diameter in inches of tail shaft} = d + \frac{P}{K}.$$

d is the diameter in inches required for the intermediate shaft of the installation; P is the diameter in inches of the screw propeller.

The value of K for ocean and cross-Channel steamers (Nos. 1 and 2 certificates, Board of Trade) is 120 when the liner is continuous, and 100 when the liners are non-continuous.

Tail shafts which are in stern tubes may have the end forward of the stern gland tapered down to a diameter at the coupling flange equal to 1·05 times that required for the intermediate shaft.

The taper of tail shafts at the propeller boss should be at the rate of 1 in 12; that is, a reduction in diameter of 1 in. to the foot.

Outboard and other intermediate shafts must be of sufficient size to avoid bending under their own weight or inertia to such an extent as to cause whipping.

The maximum unsupported length of shaft, that is the distance apart of their bearings as measured from their inner edges, shall be in accordance with the following rules:

When the maximum number of revolutions is R per minute.

For solid shafts whose diameter is d, in inches:

$$(a)\ \text{L, the length in feet,} = \text{F}\sqrt{\frac{d}{\text{R}}}.$$

For hollow shafts whose diameter is d and the bore is d_1, both in inches:

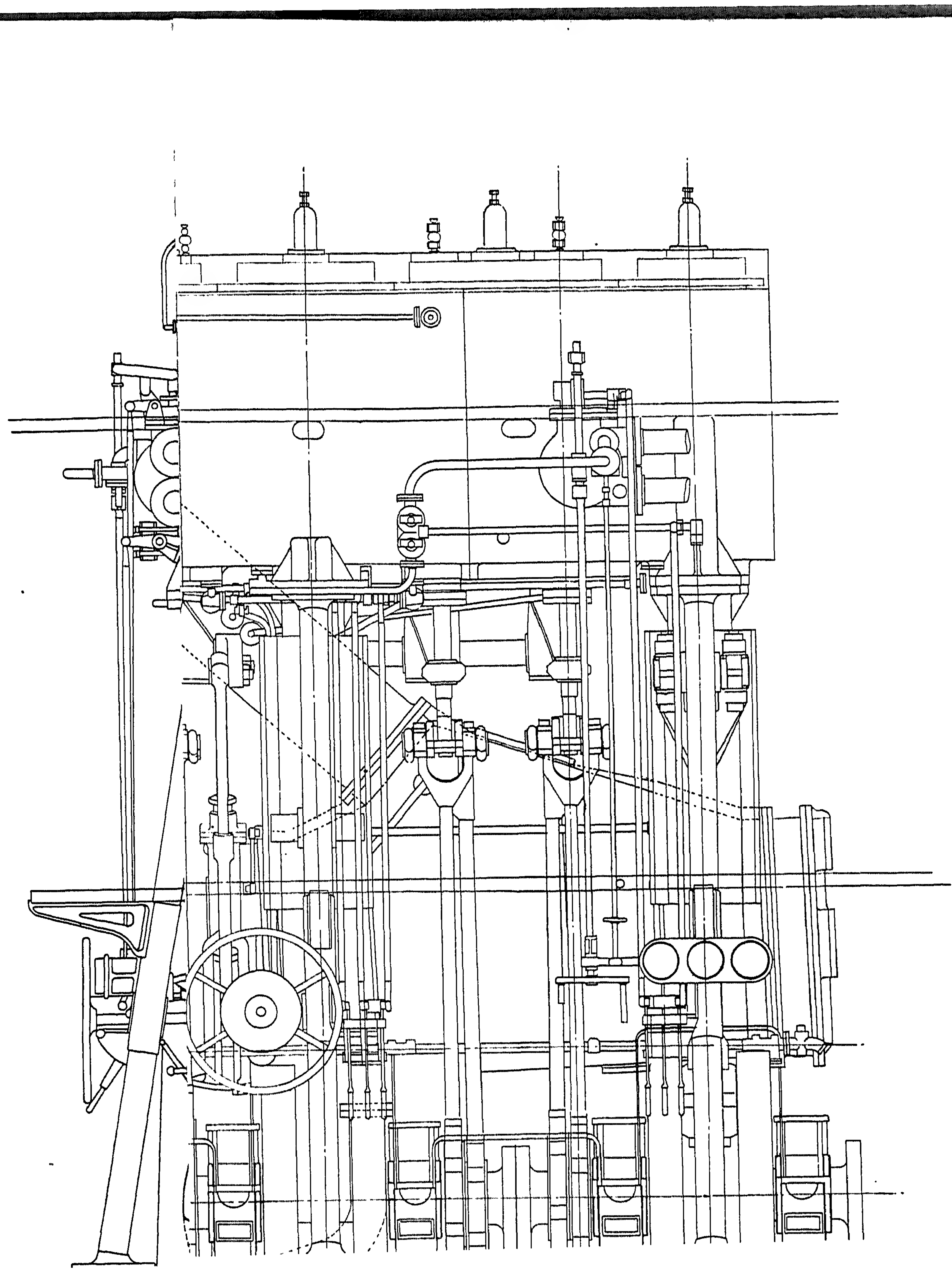
$$(b)\ \text{L, the length in feet,} = \text{F}.\sqrt[4]{\frac{d^2+d_1^2}{\text{R}^2}}.$$

For the outboard shafts of sea-going ships, F = 125.
For the inboard shafts of sea-going ships, F = 145.
For the shafts generally of ships habitually working in smooth water, F = 160.

CHAPTER III

High-speed Steam-engines

Introductory.—The high-speed enclosed self-lubricating steam-engine is substantially an English development. It came into existence in the early days of the electrical industry to meet the demand for a prime mover with a higher rotative speed than was then common, and suitable for direct coupling to dynamos which were then always belt driven, often by engines of the portable or semi-portable type. It met the needs of the situation very well, and although not so economical as the slow-speed Corliss and drop-valve engine driving through belts or ropes, it greatly reduced capital cost, not only in the engines themselves but by requiring less floor space. The development of the infant electrical industry in this country was greatly helped by the introduction of this type of steam-engine. A feature soon found to be essential to its success was that of the self-lubrication of the moving parts, which in the famous Willans engine took the form of splash lubrication.



Self-lubrication naturally led to the whole of the moving parts being enclosed, giving obvious advantages from the point of view of cleanliness, with reduced risk of grit and dirt finding their way to the wearing surfaces.

The Willans engine was of the single-acting type, and provided with a device for keeping the connecting-rod in " constant thrust ", so that there was little tendency for " knocking " at the crosshead-pin or crank-pin bearings. Consequently the wear was small.

Double-acting engines of the open type had been used for dynamo driving by makers of small naval machinery, such as was used in ship's pinnaces, &c., but the long runs at or about full power required for the generating of electricity caused rapid wear in the bearings with all the attendant inconveniences.

The double-acting type of engine was thus severely handicapped in its usefulness in generating stations and in other situations where a high rotative speed for long periods was required, but a notable invention of Mr. A. C. Pain, of the firm of Messrs. Belliss & Morcom, of Birmingham, enabled the double-acting engine not only to hold its own but to drive its single-acting rival altogether from the field. This invention, like most others of a really useful character, was very simple, and was founded upon observation of a very common incident. It had been noticed by Mr. Pain that a certain marine engine " bottom end ", which was in need of adjustment and knocking badly, always ran quietly immediately after receiving a supply of oil, but that this improved condition soon disappeared. The deduction was made that if the film of oil periodically renewed from the oil-can could be made permanent, both wear and noise would be, if not altogether prevented, much diminished. An experimental engine on this principle was constructed by Messrs. Belliss & Morcom to Mr. Pain's design, a pump being used to supply the oil to the crank-pin, crosshead-pin, guides, and main bearings, through a specially devised system of oil channels, and the result was entirely successful. Shortly afterwards, the late Mr. Alfred Morcom suggested the employment of a valveless pump of the oscillating type, and there has been practically no change in the system.

The principle has been applied to all kinds of reciprocating engines, steam and oil, also to the bearings of air compressors and other reciprocating machinery.

Considering what happens in the case of a crank-pin, for instance, it would appear that during the reversals of the direction of driving effort in the connecting-rod, causing relief of pressure on one side of the bearing, the oil film is renewed, and, as there is apparently not time for it to be squeezed out during the succeeding stroke, the rubbing surfaces never come into real contact, the oil in the clearance space acting as a cushion. The wear and noise are consequently negligible, and there are numerous examples of engines of this type which have run for many years and made millions of revolutions, and in which the wear on crank-pins and main bearings, &c., has been only just detectable by careful gauging.

It is impossible, however, to omit due recognition of the merits of the

engine invented and developed by the late Mr. Willans, or of, what is perhaps of more importance, the influence of his investigations into the actual performance of steam-engines upon the thermal design of prime movers generally. His happy inspiration to weigh the condensate from his engines at different loads and to plot the result, led to the discovery of the Willans "straight-line law", which has had far-reaching effects and has given the designer a qualitative test of performance which had been lacking. By thus giving purchasers a ready means of checking guarantees, the performance, not only of steam-engines but also of turbines, has steadily improved under the incentive of competition. Freak designs, the product of misapplied ingenuity, were thus quickly suppressed, and the rapid development of the electrical industry greatly promoted.

The Willans engine possessed many novel mechanical features, evolved to meet the combination of high rotative speed and constant thrust, but these are now mainly of historical interest, as the engine is no longer manufactured.

Each crank was driven by a complete engine, so that the consumption per indicated horse-power was for an engine of only one-third the output in the case of a three-crank engine. The multiplicity of rubbing surfaces (for in a three-crank triple engine there were twelve pistons, including one "air buffer" and nine piston valves) gave a low mechanical efficiency, and these features caused the steam consumption, when reckoned on the basis of brake horse-power, to be much higher than that of the plain double-acting triple-expansion engine with three cylinders only.

There have been many varieties of high-speed engines, but the only one surviving of importance is that developed by Messrs. Belliss & Morcom, of Birmingham, and which alone will be described in these pages. It must be understood, of course, that there are other makes of this type which have each special features of interest, examples of which will be given.

General.—High-speed engines are made in standard sizes, and because of their intimate relation with the electrical industry, the rotative speeds which have been fixed for electrical generators have been largely adopted for the engines, so that this fixed condition is the starting-point in design. The power of a high-speed engine is often given in brake horse-power and sometimes in kilowatts. The indicated horse-power is never stated. It is a figure interesting principally to the manufacturers in design.

The relations usually adopted of the powers and speeds is given in the table:

Kw.	25	50	75–100	150–200	250–400	500–750	1000
B.H.P.	38	75	112–145	220–290	365–570	720–1000	1450
Revs.	650	575–600	500–525	428–500	375	300	250

In the case of alternators, the frequency is, of course, the determining factor with regard to speed.

The indicated horse-power required depends upon the mechanical efficiency, which is very high in this class of engine, varying at full load from 90 per cent for the smaller powers and 92 per cent for medium powers to 93 or even 94 per cent for the largest. Experiments have shown that with any particular engine the amount of power absorbed by friction is practically constant at all loads. If we plot indicated horse-power against brake horse-power, we get the diagram fig. 34, which shows how the

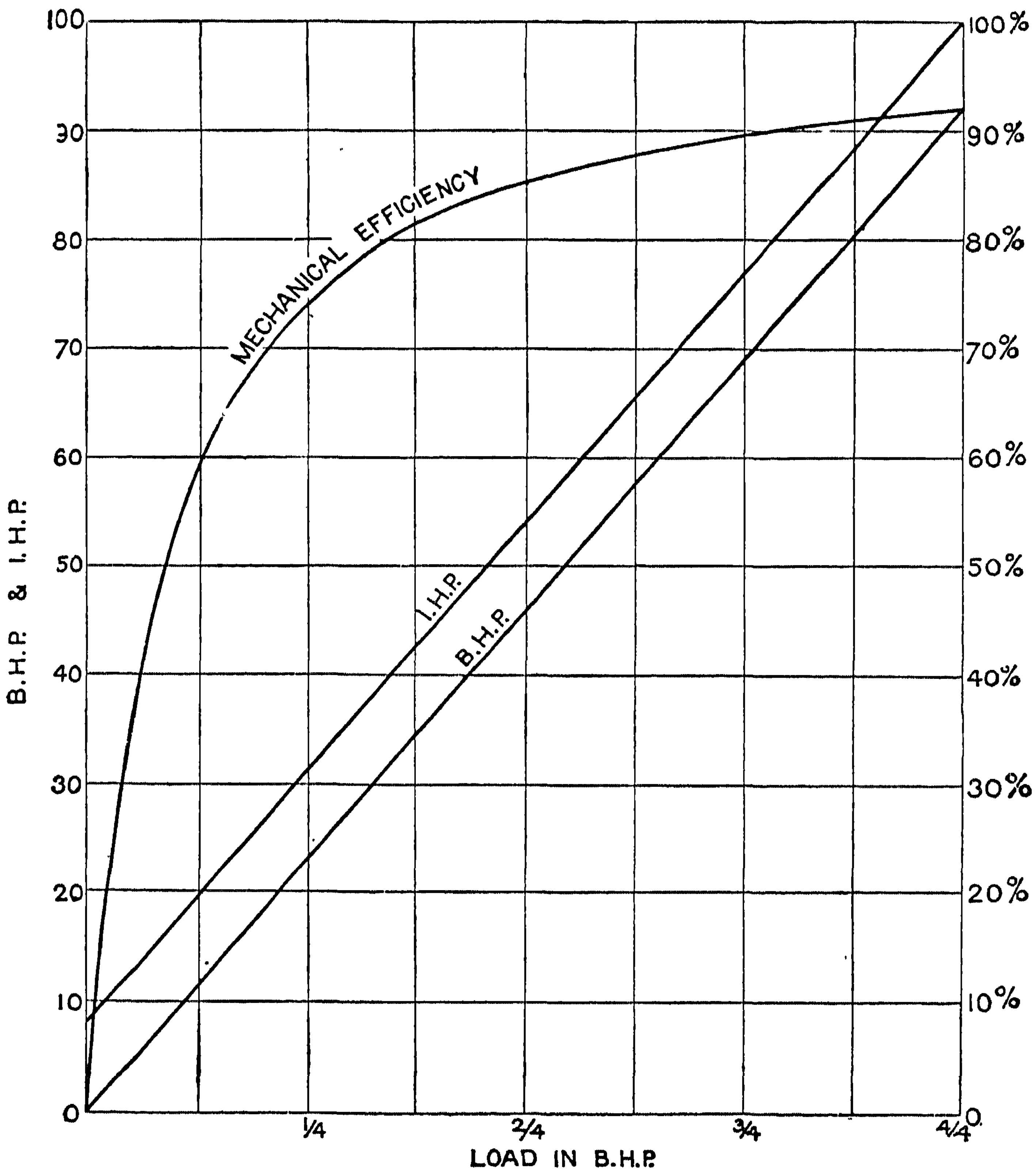


Fig. 34.—Mechanical Efficiency Curve

efficiency varies with the load on the engine. The line of indicated horse-power is parallel to the inclined brake horse-power line because of the constant difference mentioned above. The figure is drawn so that "100" represents the full load indicated horse-power.

Piston speeds vary from 500 to 750 ft. per minute for two-crank compound engines, and may reach 1000 ft. per minute for three-crank engines according to size. A three-crank engine with the cranks arranged at angles of 120° may have a higher speed than an engine with two cranks opposite to each other, the stroke being the same in each case, as the rocking couples are relatively much less in amount.

Simple engines are suitable for pressures below 80 lb. per square inch

gauge, compound engines for pressures up to 150 lb. per square inch, and triple-expansion engines for higher pressures.

In designing an engine, the size of the low-pressure cylinder is fixed first by assuming that the whole of the work is done therein by the steam expanding from the steam-chest pressure. The mean pressure which would then be developed is called the " mean pressure referred to the low-pressure cylinder " or " the mean referred pressure ". This quantity is the sum of the products of mean effective pressure by cylinder-ratio for each cylinder, the cylinder-ratio being defined as the ratio of the area of the cylinder in question to the low-pressure cylinder-area. The relation of the referred mean pressure to the initial pressure is a matter of some importance.

Formerly it was believed that the greater the number of expansions and the lower the mean pressure the higher the economy, but Willans experiments showed that it is not economical to carry the number of expansions beyond a certain point. It would obviously be wrong to reduce the pressure by expansion to a lower point than the back pressure on the other side of the piston. That is the first limiting condition even in a perfect engine. But in actual engines many conditions exist, such as wire-drawing, initial condensation, faulty release, over compression, leakage, incomplete expansion, high clearance volume, unresisted expansion between cylinders or " drop ", friction, &c., all of which reduce the thermal efficiency of the steam. Less power is obtained from a pound of steam than in a perfect engine, working, say, on the Rankine cycle between the limits existing in any particular case. If the defect of power can be regarded as an additional and unavoidable back pressure, the number of expansions permissible is reduced. The better the engine, the less will be the effect of the disturbing conditions referred to above, and the greater therefore the number of expansions and the lower the mean referred pressure to give maximum economy.

Willans gave certain figures based upon the results of his experiments, but they were obtained from an engine of small size and of exceedingly special design, and would not necessarily apply to engines of greatly different types in which the effect of the factors mentioned above would all be much modified, even in different makes of the same type of engine.

For these reasons, the questions of the relation of the most economical " mean referred " pressure to initial pressures, cylinder ratios, &c., are dependent entirely on experience, and the results of tests of particular designs of engines. In the cases of triple-expansion engines with initial pressures of 150 to 200 lb. per square inch, the ratio low-pressure cylinder/high-pressure cylinder may be from $4\frac{1}{2}$ to 7 with mean referred pressures ranging from 34 to 44 lb. per square inch condensing. The intermediate pressure-cylinder is usually made a mean proportional between the two. For compound condensing engines with initial pressures from 100 to 150 lb. per square inch, the cylinder-ratio may be from 3 to $4\frac{1}{2}$ with mean referred pressures from 32 to 40; when non-condensing, the ratio may be from 2 to 3 with mean referred pressure from 33 to 44. It is found

that a departure in either direction from these mean pressures of a few pounds per square inch does not make a very considerable difference in steam consumption.

This enables the size of the low-pressure cylinder to be fixed. The diameter of the cylinder is usually about 2·2 to 2·4 times the stroke for triple engines and 1·9 to 2·2 for compounds. The size of the high-pressure cylinder is governed to some extent by the consideration that with valve gear driven by eccentrics in the usual way an earlier cut-off than 0·55 or 0·5 is not very practicable because of the great valve travel required. This

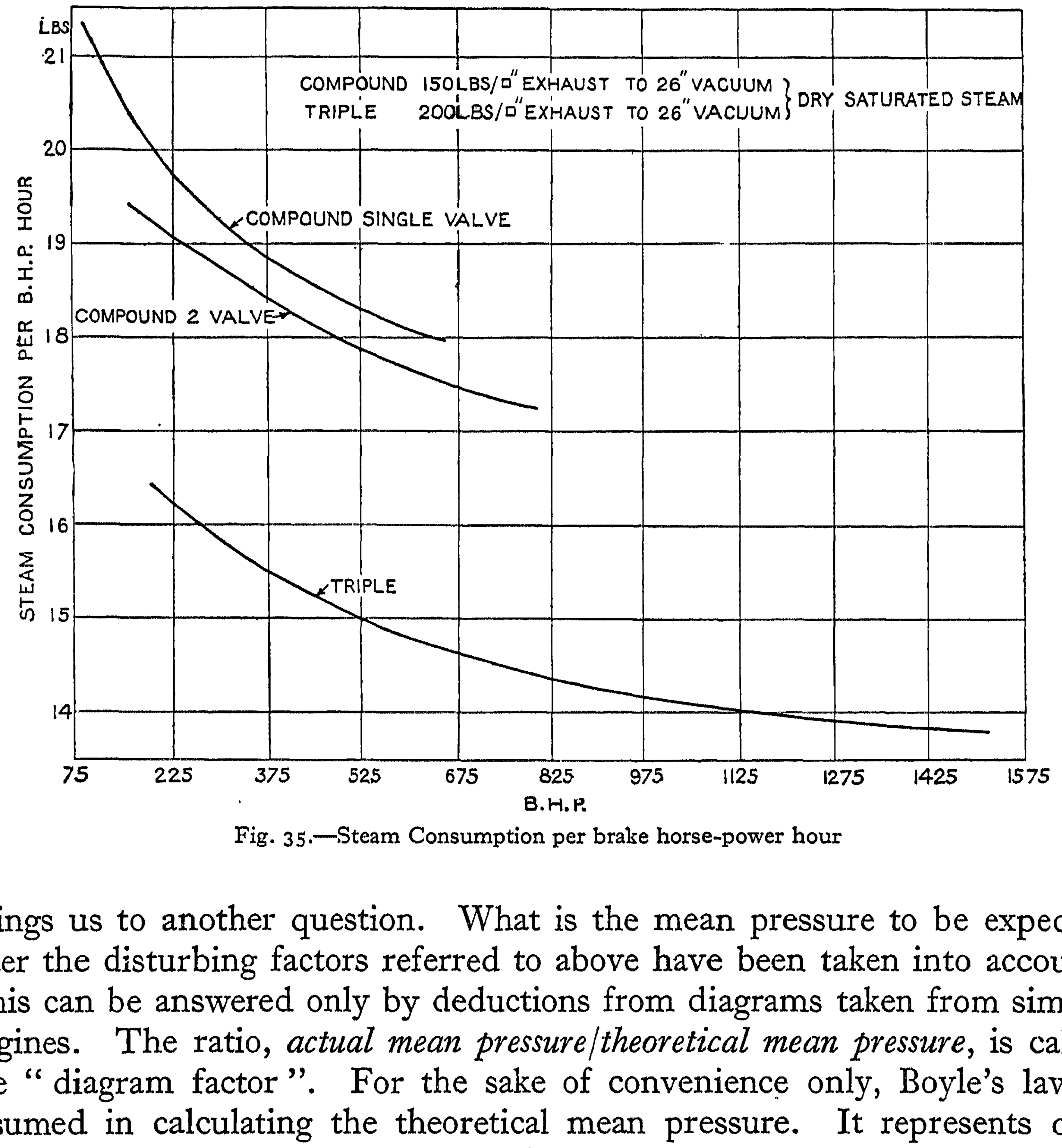


Fig. 35.—Steam Consumption per brake horse-power hour

brings us to another question. What is the mean pressure to be expected after the disturbing factors referred to above have been taken into account? This can be answered only by deductions from diagrams taken from similar engines. The ratio, *actual mean pressure/theoretical mean pressure*, is called the "diagram factor". For the sake of convenience only, Boyle's law is assumed in calculating the theoretical mean pressure. It represents only approximately the real relations of the pressure and volume of expanding steam, but is sufficiently near the truth to make comparisons of similar conditions fairly trustworthy. The diagram factor varies greatly in different engines. For a triple-expansion engine it may vary from 0·55 to 0·62 or even 0·65 in cases where the admission line has been well maintained. For compound engines it may be from 0·6 to 0·65 and for simple engines 0·6 to 0·72, depending upon whether the cut-off is early or not. If early, there would probably be considerable throttling past the valve. Obviously the pressure

in the steam chest should be taken for the upper limit, as it would not be fair to debit the engine with losses in the steam pipes and stop valves, but losses through the ports are affected by the design, and should be charged against the engine. A drop of 10 to 15 lb. should be allowed between the boiler and the engine steam chest. For the lower limit a reasonable back pressure above the condenser or the atmosphere should be assumed for inevitable losses in the exhaust ports, say 1 to 2 lb. for non-condensing engines and 2 to 4 lb. for condensing engines, and as these losses are affected by port and valve areas, as in the case of the high-pressure admission, high back pressures also should be regarded as an avoidable loss.

Cards from actual tests should be preserved, and the results, such as the drop from high-pressure steam chest to cylinder at admission, drop at the point of cut-off, and difference between actual back pressure on low-pressure cylinder, and pressure in exhaust pipe taken as near to the engine as possible, should be noted for future use.

The diagram factor having been obtained or assumed, it is then possible to fix the particulars of the high-pressure cylinder.

In order to obtain the diameter of cylinder, the cut-off must be fixed, bearing in mind the limitation imposed by ordinary valve gears previously referred to

If P_m = chosen mean referred pressure;
P_c = steam-chest pressure;
P_b = back pressure;
R = number of expansions = ratio, volume low-pressure cylinder/volume high-pressure cylinder at cut-off;

$$\text{then } P_m = \frac{P_c(1 + \log_e R)}{R} - P_b.$$

$$\therefore \frac{P_m + P_b}{P_c} = \frac{1 + \log_e R}{R}.$$

Values of $\frac{1 + \log_e R}{R}$ for varying values of R have been tabulated, and as the value of $\frac{P_m + P_b}{P_c}$ is known for any given case, it is easy to find R from the table. If the cut-off in the high-pressure cylinder be decided and expressed as a fraction r, then the area of the high-pressure cylinder = area of low-pressure cylinder/Rr. An equal distribution of power in the various cylinders is usually aimed at. To fix the cut-offs in the other cylinders from assumed clearance volumes, &c., from first principles is an exceedingly tedious and complicated process, and in practice reliable indicator cards from similar engines are carefully analysed.

Fig. 35 gives a diagram showing the full-load steam consumption to be expected from compound and triple-expansion engines of various sizes working with dry, i.e. saturated steam, and fig. 36 is a diagram showing the steam consumptions for the various loads for a triple-expansion engine of

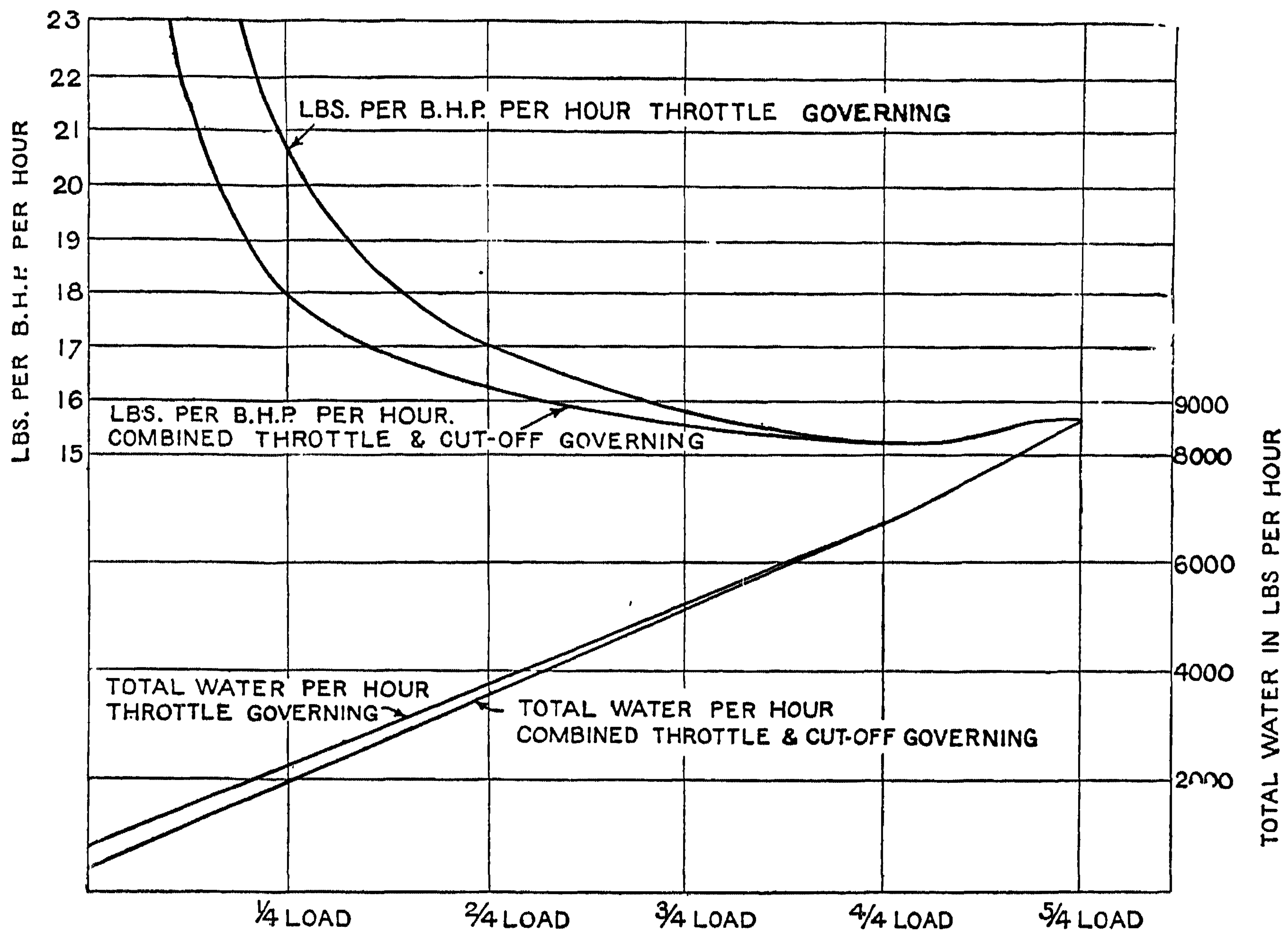


Fig. 36.—Triple 440 b.h.p. Steam pressure, 200 lb. per square inch exhaust to 26 in. vacuum

440 b.h.p. The figure shows clearly the difference in the consumptions between throttle governing and governing by " cut-off ". They are based on

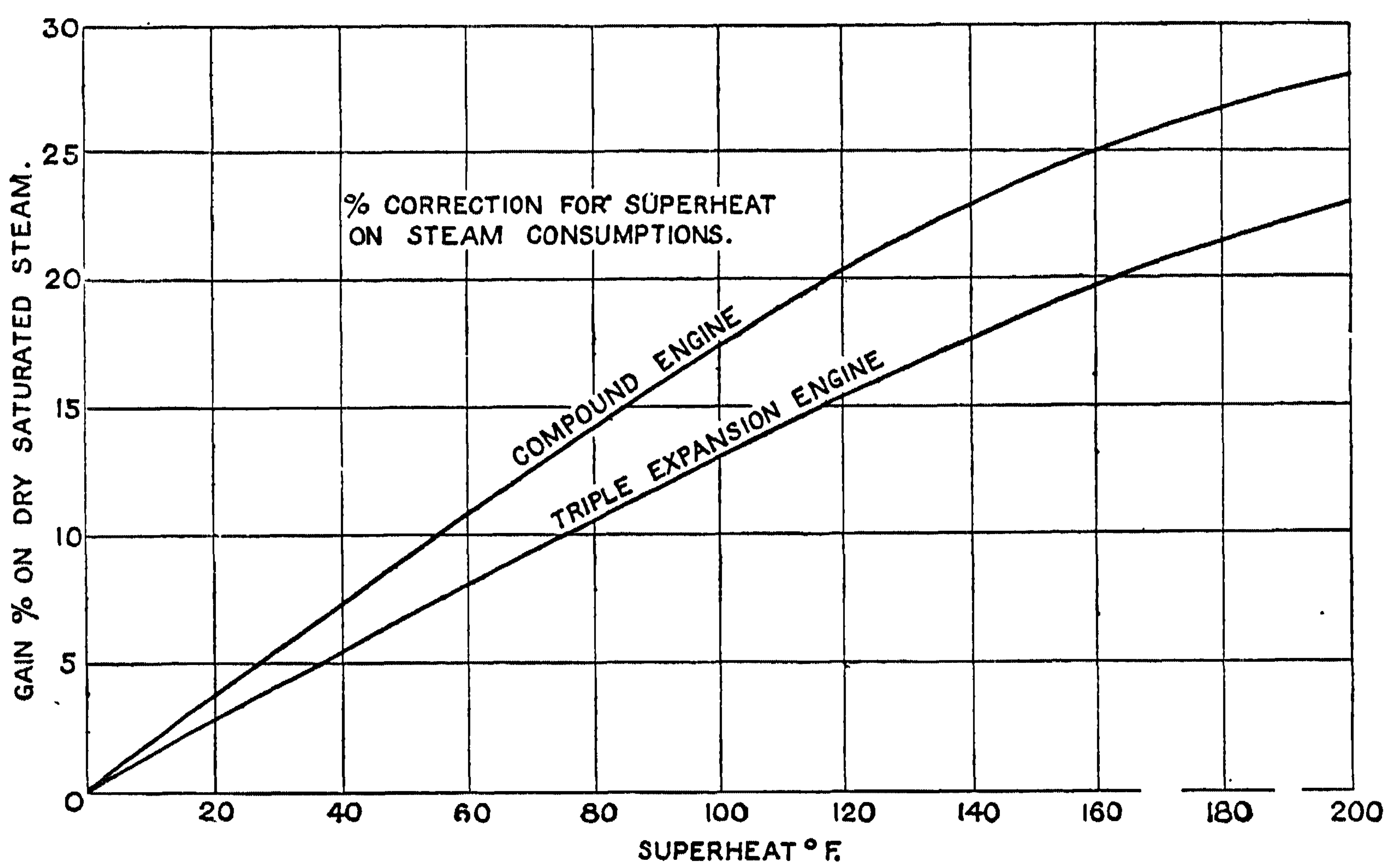


Fig. 37

actual test figures. Fig. 37 shows the corrections for superheat usually adopted.

There are also given two inclined lines which show the total water per

hour at the different loads (fig. 36). This method of plotting the total water consumption was introduced by Willans, and has been of much service to engineers. It will be noticed that when the brake horse-power is zero, that is, when the engine is running " light ", a certain amount of steam is being used.

Bedplate and Oil Pump.—The chief structural function of the bedplate is to support the crank-shaft bearings and to act as the base upon which the engine is erected.

In high-speed enclosed engines it also serves as an oil tank, and the oil pump is fixed inside on the bottom. As in the case of the crank-case, the thickness of metal is usually decided by manufacturing considerations, and it may have the same thickness, or in large engines may be a little thinner. Its cellular shape makes it exceedingly strong as a structure. In very large engines the bedplate is sometimes split longitudinally for convenience in manufacture, but this is not to be recommended, as trouble may be caused by oil leaking through the joint, causing the concrete to perish. There is no cure for this except taking down and re-erecting the whole engine and cutting away the concrete which has perished. This is a serious undertaking. For the same reason, and in the interest of cleanliness, a groove or shallow well should be cast all round the bottom edge of the bedplate to catch any oil that may drip down.

On the top surface there is a thick facing strip for the crank-case joint, which helps also to stiffen the section in that neighbourhood. Doors are provided to give access to the oil pumps and strainers.

The bearing shells are always of cast iron lined with white metal and may have a thickness of $\frac{d}{10} + 0{\cdot}2$, where d is the diameter of the shaft in inches. Bronze shells are not necessary. With cast iron there is no possibility of trouble occurring by the shells closing on the shaft, owing to the difference of the coefficients of expansion of bronze and cast iron, as the metal, of course, warms up when the engine is running.

The bearing caps may be of cast iron, but some makers prefer to use cast steel. The caps act as a beam, supported at each end at the centres of the studs. The bending moment may be taken as $\frac{WL}{8}$, when L in. is the distance between stud centres, and the stress should not exceed 1000 to 1200 lb. per square inch for cast iron and 6000 lb. per square inch for cast steel. In taking out these stresses the full boiler pressure should be assumed on the high-pressure piston for the high-pressure bearing caps, due allowance being made for the distance between the centre of the crank-pins and the centres of the respective bearings. For the other bearing caps the greatest load due to working conditions including overload should be taken from indicator diagrams of similar engines, regard being given to the sequence of cranks, in triple engines especially, and the total load that may come upon one cap should be taken. Allowance should be made for the hole for the oil pipe in the cap. Studs are invariably used, and the stress

should not exceed 3500 lb. per square inch for small engine, and 5000 lb. per square inch for larger engines.

In passing, it may be remarked that in all cases such screwed parts as the studs, also connecting-rod and crosshead bolts, usually have fine threads, and not the standard Whitworth thread.

The bearing surface should be such that the pressure per square inch does not exceed 250 lb. per square inch of projected area, but with the ordinary design of high-speed engine and arrangement of cylinders the length and diameter of the crank-shaft is such that if nearly all of the space not taken up by the eccentrics and cranks is utilized for bearings, the pressure per unit area of surface is usually satisfactorily low. The outer low-pressure journal is usually made 10 per cent larger in diameter and about 33 per cent longer than the other, in order to take the weight of the fly-wheel. The pressure due to the weight of the fly-wheel alone should not exceed 100 lb. per square inch.

The height of the bedplate should be such that the connecting-rod bolt heads do not dip into the oil and so churn it up. The bottom floor of the bedplate should slope downwards towards the part where the oil pump is fixed, so that the oil may drain back to the pump strainers.

The bedplate should be designed in such a way that no part projects below the surface supported by the foundation, for convenience in erecting and grouting. The under surface should, of course, be machined. The holding-down bolts should be well distributed.

The oil pump is, of course, mounted in the lowest part of the bedplate. It is always of a simple oscillating type without valves or stuffing box, as originally introduced by Belliss and Morcom, and has no loose parts whatever, consisting merely of a barrel with trunnions, or their equivalent, the bracket in which the trunnions oscillate and which contains the suction and delivery connections, and the ports to the trunnions and the plunger. All the parts are of cast iron, except in the smallest sizes, when they may be of gun-metal. The plunger is driven by means of a pin engaging with a lug on one of the eccentric straps.

In large engines two pumps are fitted, each capable of serving the engine alone. This enables the strainers of each pump to be taken out and cleaned whilst the engine is running, a shut-off cock between the strainer and the pump suction being provided.

The connecting pipes to the bearings and other parts to be lubricated should be of iron or steel. Copper, although a convenient material to work, should not be used. It crystallizes and ultimately fractures, through the constant vibration to which these pipes are subjected.

A sump is provided in the bedplate to receive the drainings from the distance pieces of the mixture of oil brought up by the piston-rod and water from the glands. An ingenious separating device was introduced by Messrs. Belliss & Morcom, which depends for its action upon the difference of the densities of oil and water. Two vertical pipes are fitted in the sump in such a way that the level of the open end of one pipe is slightly higher than

that of the other. The mixture of oil and water settles down in the sump, and the oil slowly separates and floats to the top, whence it escapes by the higher pipe and is led into the bedplate to be used over again. The water escapes by the other pipe, and is led away outside the engine.

Cylinders and Distance Pieces.—In the case of triple-expansion engines, the cylinders are usually separate castings, but some of the smaller compound two-cylinder engines, say up to 200 kw., often have the cylinders cast in one piece.

Sometimes a liner is fitted to the high-pressure cylinder. This is not usual, but with modern high pressures and temperatures it is advisable. Much damage may be done by scoring, due to the temporary failure of lubrication or the use of a poor quality of cylinder oil. In such cases a liner is a great convenience. It is also convenient from the point of view of design, as it enables various conditions of initial steam pressure to be met, without the main pattern requiring much modification. There are no very definite rules for the thickness of steam cylinders, most formulæ being of the empirical type. Usually the barrel stresses are kept low and an allowance made for reboring, but the possibility of inequality in the thickness of the casting and other considerations are also taken into account. The barrel stresses due to the steam pressure should not exceed 2400 lb. per square inch for high-pressure cylinders. For intermediate and low-pressure cylinders the stresses may be 2000 lb. and 1000 lb. per square inch respectively. The steam pressures assumed for these calculations are boiler pressures from 180 to 210 lb. per square inch for high-pressure cylinders, 100 to 120 lb. per square inch for intermediate-pressure cylinders, and 30 to 40 lb. for low-pressure cylinders. For compound engines the boiler pressure should be taken for high-pressure cylinders. This is usually lower than in the case of triple-expansion engines, so that the pressure for the low-pressure cylinders of compound engines may be taken as about the same as for triple-expansion engines. In each case an allowance should be made for reboring, say $\frac{1}{4}$ to $\frac{3}{8}$ in. according to size, but many high-speed engines have worked for very long periods without requiring this operation.

The design of the cylinder should be kept as simple as possible, flat surfaces in belts and passages being avoided. Bosses for drain cocks and relief valves should not be larger than necessary, in order to avoid local sponginess in the casting. All flanges and branches should be joined to the main body by generous fillets. The cylinder or liner should be counter-bored, finishing with a taper at the level of the lower edge of the top port to facilitate the introduction of the complete piston into the cylinder and to allow the upper piston rings to overtravel the edge of the counter-bore. Sometimes a groove is made in the cylinder wall near the bottom to allow the lower edge of the bottom piston ring to overtravel in the same way.

The bottom cover is usually incorporated with the distance piece which connects the cylinder and the crank-case, so that the cylinder itself is a plain casting. Some makers cast the distance piece with the cylinder, but this is to a very great extent a question of manufacture depending upon the

class of tool used for boring, &c. The stresses in the studs at the bottom of the thread are from 4000 lb. per square inch for small studs to 7000 lb. per square inch for large studs. The diameter of the studs is governed by the consideration that in the case of high-pressure cylinders the pitch should not be more than 3 to $3\frac{1}{2}$ times the diameter of the studs. In the case of low-pressure cylinders the pitch may be as much as 7 to 8 times the diameter. Great care must be taken to provide bosses where necessary in ports and passages, to prevent the stud holes being drilled through into a steam space. This point sometimes is overlooked. The flanges and covers near the joint may have a thickness equal to the diameter of the studs plus $\frac{1}{8}$ in. in the case of small studs, and diameter plus $\frac{1}{2}$ in. for larger studs. All the studs should have the same diameter and the covers the same thickness for all cylinders.

The top covers are cast to suit the form of the pistons and are well ribbed for strength, the number of the ribs varying with the diameter of the cover and the steam pressure. The thickness of the cylinder covers and the ribs may be 0·7 to 0·8 of the thickness of the cylinder walls. A recess is formed in the cover for the piston-rod nut, and a circular rim or projection is often provided, which extends as high as the top of the recess, to accommodate the lagging plates for the cover.

When designing a cylinder, provision for the attachment of the lagging should not be forgotten, although the flanges themselves are usually sufficient for the purpose.

The area of cross section of the ports should be based upon the speed of the exhaust steam, which should not exceed if possible 100 ft. per second for high-pressure cylinders and 120 ft. per second for low-pressure cylinders. In the case of the latter, especially in large engines where piston speeds are high, it is often necessary to increase the speed of the steam, but then sometimes two piston valves are used, the spindles being connected to a common crosshead in the crank-case driven by one eccentric.

The distance pieces should be carefully designed, as they are subjected to heavy shock when water gets into the cylinders. The stresses should be kept low, not higher than 800 to 1000 lb. per square inch. The top of the distance piece forming the bottom cylinder cover is usually coned to suit the piston, and is therefore of an inherently strong form. In the case of the high-pressure cylinder it is sometimes flat to suit the shape of the piston, but the presence of the stuffing box makes it possible to provide deep ribs in all cylinders. The studs which attach the distance piece to the crank-case can be less in number than those used for the cylinder, as there is no joint to be kept tight, but they will, of course, have a proportionately greater diameter. The bottom distance-piece flange should therefore be made thicker than the cylinder covers, to allow for the greater distance bridged by the studs. The distance piece is usually registered into the circular hole in the top of the crank-case, but some makers prefer to use dowel pins instead, thus giving means of adjustment which may be useful when erecting the engine in the shops.

Metallic packing is invariably used. In the case of the high-pressure cylinder it is usually of the double type. This detail is never made by the engine builders themselves, but is supplied by firms who specialize in its manufacture.

To prevent oil being carried out of the crank-case by the piston-rod, a wiper gland is provided, fixed in a diaphragm in the distance piece in order that oil may not be sucked into the low-pressure cylinder by the vacuum. It is necessary that the distance pieces shall have height sufficient to prevent any part of the piston-rod which comes in contact with oil entering the low-pressure gland. Oil from the wiper glands and water from the cylinder glands collect in the bottom of the distance piece, and the mixture is drained away to a common pipe fixed inside the crank-case, and thence to a pocket or sump in the bedplate where it is dealt with by a special form of separator, to be described later.

Frames.—In good-class work the frames and the bedplates, even in the smallest sizes, are always separate castings. It may be remarked that the former is often referred to as the crank-case. Its chief function is, of course, to connect the cylinders to the bedplate through the distance piece, but it also encloses the working parts and so prevents oil being splashed about. The material is subject to tensile strains during the downward movement of the piston. The thickness of the metal is determined to a great extent by foundry considerations, so that there is always ample material to resist tensile strains. Further, if the casting is too thin but yet of ample strength, an unpleasant "drumming" noise is caused when the engine is running. For this reason, apart from considerations of strength, all flat surfaces should be well ribbed. The horizontal flat top of the crank-case which supports the distance pieces should be thickened to take the studs for the distance pieces, and well reinforced inside by transverse ribs in the neighbourhood of the distance-piece stud circles. These ribs may be carried down each side of the crank-case inside, gradually decreasing in depth towards the bottom. The whole of the metal in the section of the crank-case between the ribs, including the depth of the ribs themselves, may then be regarded as resisting the steam forces on the piston.

Some triple-expansion engines have three doors, one opposite each line. This perhaps adds to the convenience of inspecting, but the metal of the crank-case above the doors is subject to bending stress, and the section and depth of the metal should be sufficient to make this very low. It is better to have the doors between the crosshead guides, as it is then possible to have a continuous section right down to the bottom flange of the crank-case.

The bottom flange for attaching the crank-case to the bedplate is usually heavy, in order that there may be no spring between the studs, which may be a little greater in diameter than the studs for the cylinders and pitched 7 to 8 diameters apart. For the largest engines the crank-case is often divided into an upper and a lower portion for convenience in manufacture and transport.

Valves and Liners.—Piston valves are now exclusively used for high-speed engines. In many cases they are quite solid and have no packing rings whatever. The valves are finished to size in a grinding machine with several water grooves turned in the faces. The usual practice, however, is for the active part of the valve to be made as a floating ring with plain faces held in position by light end rings secured by studs in the body of the valve. The body casting should be as simple as possible in order to avoid the distorting effect of ribs. Some manufacturers make the body of the valve a plain open cylinder without any ribs whatever, the floating rings at the end being kept in position by strong caps which are in turn secured between the collar on the valve rod and the nuts at the end. It is better, however, to secure the end caps to the valve body by studs, as the parts of the other design have to be inserted or taken out of the valve chest separately. In cases where Ramsbottom or other forms of expanding rings are fitted, it

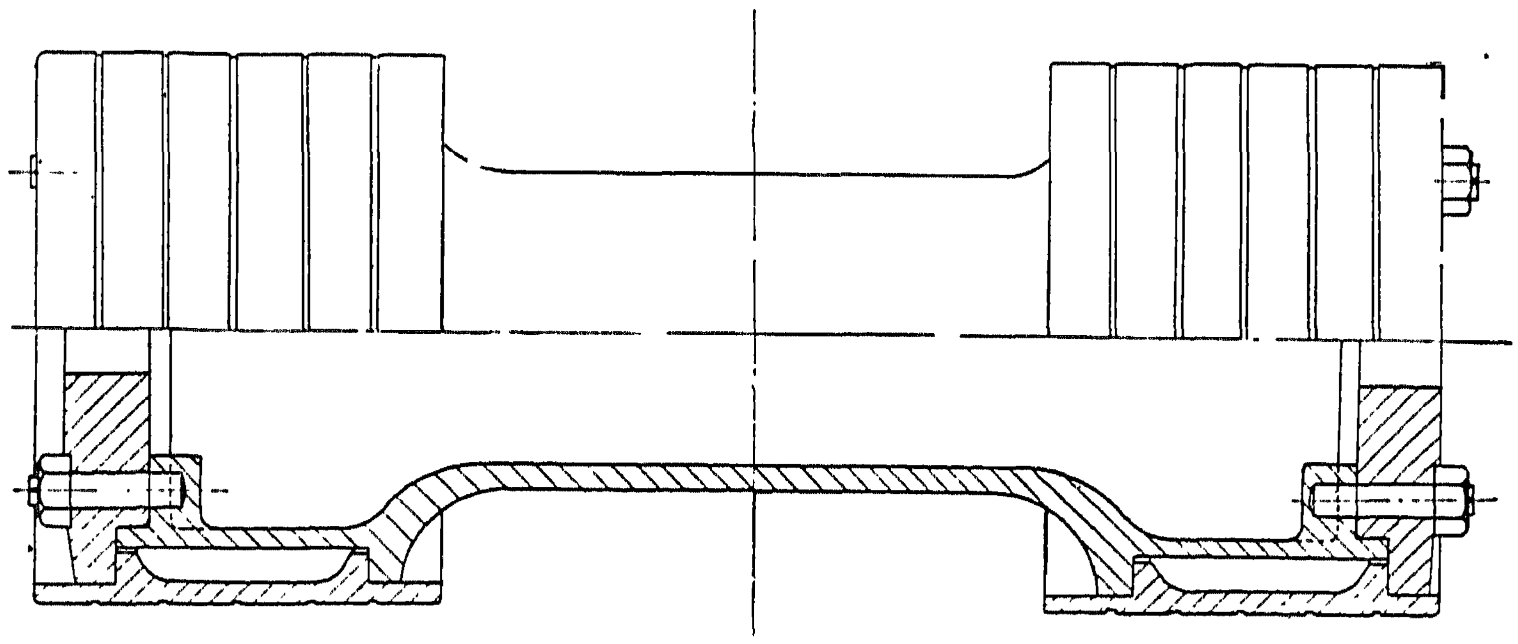


Fig. 38.—Piston Valve for High-speed Engine

is necessary to use liners in the valve chest. These are pierced with holes in order to communicate with the steam port. The bars between the openings should be placed at an angle to the axis in order to prevent local wear on the valve rings. The presence of a liner necessarily causes a certain drop in the initial pressure in the cylinder, and for this reason many designers prefer to dispense with liners, using, of course, solid valves or floating solid rings.

In the early days of the double-acting high-speed engine a single valve to serve both high-pressure and low-pressure cylinders in compound engines was arranged in a chest between them. It had two portions threaded on a common valve spindle driven from one eccentric. The high-pressure end of the valve had usually "inside" admission and the low-pressure end "outside" admission, the two cranks being, of course, opposite to each other. The valves were hollow, and the exhaust steam from the outer high-pressure port passed through both portions of the valve to the outer low-pressure port. This design is not now so commonly used, as it is found much better from the point of view of economy for each cylinder to have its own valve, the exhaust passing from the high-pressure to the low-pressure through an external pipe which is, of course, well lagged to prevent condensation. Messrs. Belliss & Morcom have introduced a type of engine having a valve for each cylinder,

the two valve chests being removed from the central plane of the engine and inclined to each other, their axes meeting at the centre of the crank-shaft.

Fig. 38 illustrates a common type of piston valve, which can be used when the steam has a fair degree of superheat.

Pistons.—Although the types of piston and packings that have been used are exceedingly numerous, most of them have become obsolete, and it would be useless to describe them. Experience has caused the design to settle down in favour of a very simple type which is characterized by lightness, simplicity, and the absence of loose parts which may become detached during working and so cause damage. The design of a piston is governed by the consideration that the reciprocating parts should be as light as possible in order to minimize inertia effects, and the low-pressure piston is the starting-point. It is invariably of the conical type, fig. 39, either of cast or pressed steel. As it is important that the reciprocating weights should be the same for each cylinder for the purpose of giving good balancing, the high-pressure and the low-pressure pistons are made of cast iron. This ensures ample strength, and it is not necessary to consider this point in their case. It may be noted that in the case of triple-expansion engines it is hardly ever possible to get sufficient weight in the high-pressure piston. The strength of the low-pressure piston alone is therefore to be considered. There are no very satisfactory formulæ available, and the designer is guided by experience in this matter. Satisfactory proportions are $T = \frac{D}{35} + 0{\cdot}41$ and $T_1 = 0{\cdot}44T$.

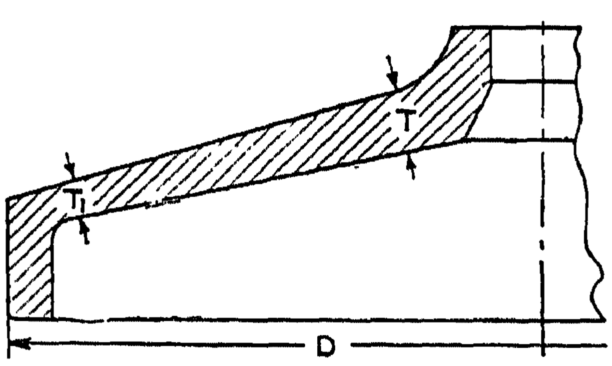


Fig. 39.—Typical Section of a Piston

The type of piston packings is first decided upon, designs involving great weight being rejected, and the thickness of metal in the wall behind the ring is fixed. The bottom of the piston is then given a slope which varies from 1 in 7 to 1 in 8. The pistons for the other cylinders are made exactly alike with regard to height of piston-rod attachment, so that the rods for all cylinders may be identical, in order to reduce the number of spares, &c.

The packing for the low-pressure piston presents little difficulty, as, owing to the low pressure and temperature, a very simple type of ring, reinforced behind by light springs, suffices. Rings of the Ramsbottom type are usually quite satisfactory. This also applies to the intermediate cylinder of triple-expansion engines; but the piston rings of the high-pressure cylinders for both triple-expansion and compound engines, in which high pressure may be accompanied by a high temperature, may give a great amount of trouble, owing to the impossibility of preventing the steam getting behind the rings and forcing them on to the cylinder walls with an intensity equal to a high percentage of the pressure in the cylinder at the moment. This pressure causes the lubricant to be scraped away, allowing metallic contact, and much damage may be done to the bore of the cylinder. For this reason some type of restrained ring is commonly used in high-

pressure cylinders, and this involves the use of a junk ring as the restraining member is, of course, not split. There are many varieties of piston rings manufactured by private firms or specialists, and these are sometimes specified by users because of the special advantages that are claimed for them. One of the best known for high-speed engine work is the "Rowan", of which details are given.

When junk rings are used, special precautions must be taken to prevent the possibility of the set screws slacking back and coming adrift. In all cases tapped holes should be provided in the piston boss, into which eye bolts can be screwed for the purpose of withdrawing the piston from the cylinder.

Piston-rod and Crosshead, Slippers and Guides.—The usual method of fixing the piston to the rod is by means of a nut screwed on to the end of the latter. The nut is of the castle type, grooves being milled across the face to take a split pin. The part of the rod inside the piston boss consists of a parallel part having the same diameter as the top of the thread, and a taper part. It is bad practice to make the taper too small. The rod is then difficult to extract from the piston, and in the event of water getting into the cylinder the rod may burst the piston by wedging action. The taper is usually made 1 in 4 on the diameter, and this is satisfactory for all types of engine. The tensile stress allowed upon the section at the bottom of the thread is 5000 lb. per square inch for small engines, and 7000 lb. for large engines, the load being taken at full boiler pressure on the high-pressure piston. Some makers use a high-tensile steel for piston-rods, and the stress may then be somewhat greater. It should not be forgotten that such parts as piston-rods and crosshead and connecting-rod bolts are nicked by the screwed part and have an inherent tendency to failure there. The stress on the body of the rod is usually very low, as the diameter is much greater than the screwed part, owing to the taper of the part in the piston and the shoulder left upon the rod. In addition, there is an allowance of $\frac{1}{8}$ to $\frac{1}{4}$ in. for re-turning. High-speed engine piston-rods are short compared with their diameter, and this condition, combined with the low stress, makes failure by crippling unlikely.

The combination of crosshead, guide, and connecting-rod top end have given rise to many variations in design with all kinds of screw, wedge, and cotter adjustments and fastenings, but generally only two types are now used, both being taken almost unchanged from marine engine practice. This may be accounted for by the fact that the high-speed engine was to a great extent developed by firms who had had previous experience in naval and marine work, where the types referred to have been found quite reliable and satisfactory.

In one type the crosshead pin is fixed in the jaws of the connecting-rod and partakes of its angular movements, the bearing remaining fixed and forming part of the crosshead. In the other type the bearings are carried by the jaws of the connecting-rod, the gudgeons being solid with the crosshead body. Both types necessitate a forked connecting-rod end.

In the type first mentioned the crosshead body and the piston-rod are in one solid forging, or there is a palm on the rod, which forms a base for the part in which the bearings are seated. Both these designs necessitate that the rod be withdrawn downwards into the crank-case, but this may be an advantage, especially in small engines used on board ship where head room may be restricted.

The slippers are of cast iron and are secured to the crosshead forging by screws with countersunk cheese-heads. A tongue is formed on the outer surface of the crosshead, which fits in a recess or groove turned in the slipper. This serves to locate the slippers and takes the forces due to inertia and friction. The slippers are secured to the crosshead by two or four phosphor-bronze screws with countersunk cheese-heads. This design is illustrated by fig 40.

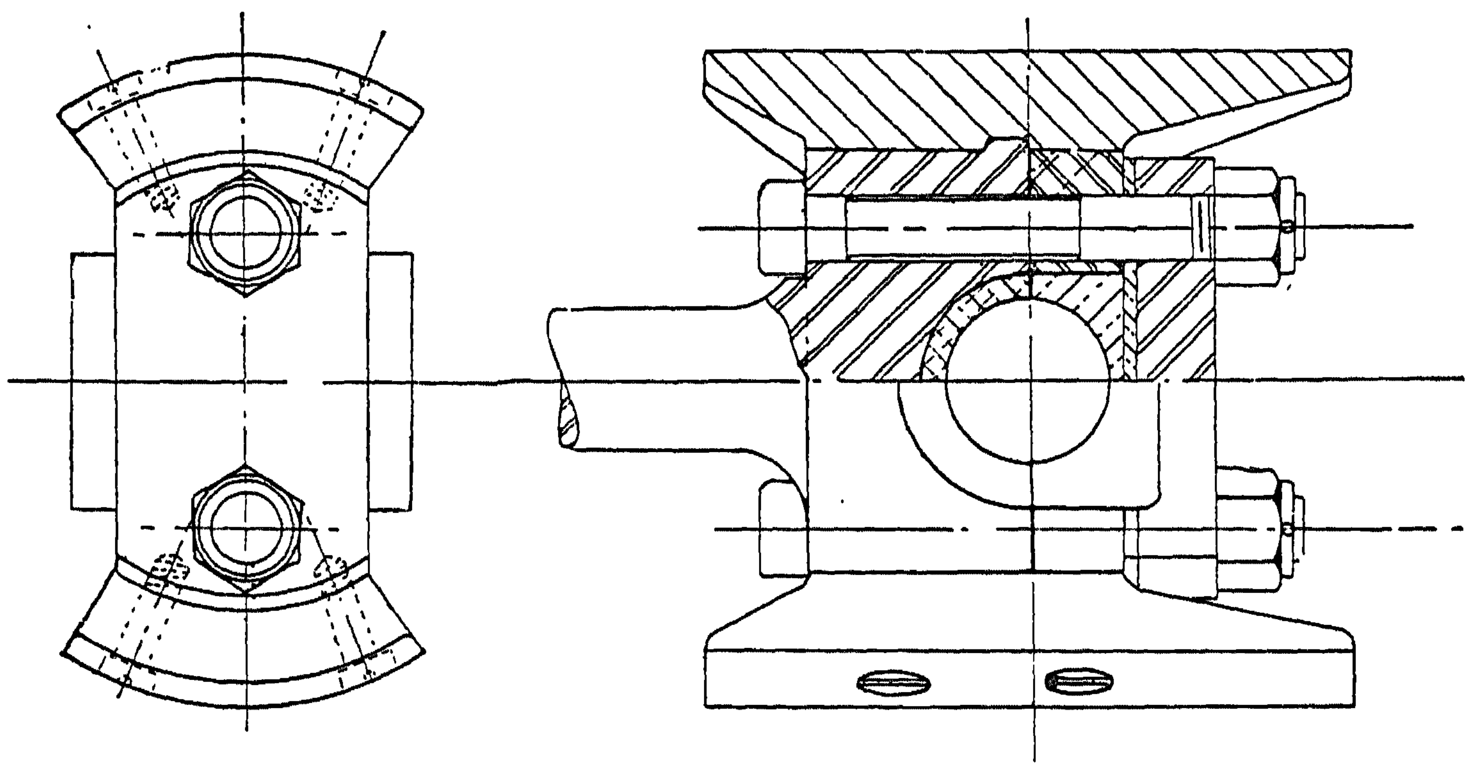


Fig. 40.—Crosshead for High-speed Engine

Another variety to suit the same type of connecting-rod is shown by fig. 41 In this case the crosshead is secured to the piston-rod by a palm upon the latter. The crosshead body is of cast steel, the slipper being cast with it. The whole structure is held together by the two bolts. In this case a flat guide is used, fitted with guide strips in the usual way. Sometimes the face of the slipper is lined with white metal, but experience has proved that the low bearing pressure and copious forced lubrication render the white metal unnecessary. The maximum bearing pressure due to the piston load at the point when the angularity of the rod is greatest, multiplied by the ratio of the connecting-rod to the crank, should not exceed 35 to 50 lb. per square inch over the whole surface of the slipper.

The crosshead bolts may be given a stress of 4000 to 5000 lb. per square inch at the bottom of the thread, and should be turned down in the body to that diameter to give more resilience, collars being left of the full diameter at each end and opposite the joint in the brasses. It is most important that a good fillet should be left under the head of the bolt.

The brasses should be made of phosphor bronze or hard gun-metal, and may have a thickness at the crown of $0{\cdot}2d + \frac{1}{8}$ in., where d in. is the

diameter of the crosshead pin. The thickness at the sides may be somewhat reduced. White metal is never used in this position, as it would not stand the hammering action during working. The brasses should butt at the joints, no liners being used. The cap is made of mild steel, and should be calculated with a bending moment of $\frac{PL}{8}$, where P lb. is the total piston pressure and L in. is the distance between centres of bolts. A stress of 8000 lb. per square inch may be allowed, but often the thickness is made about equal to or a little more than the bolt diameter.

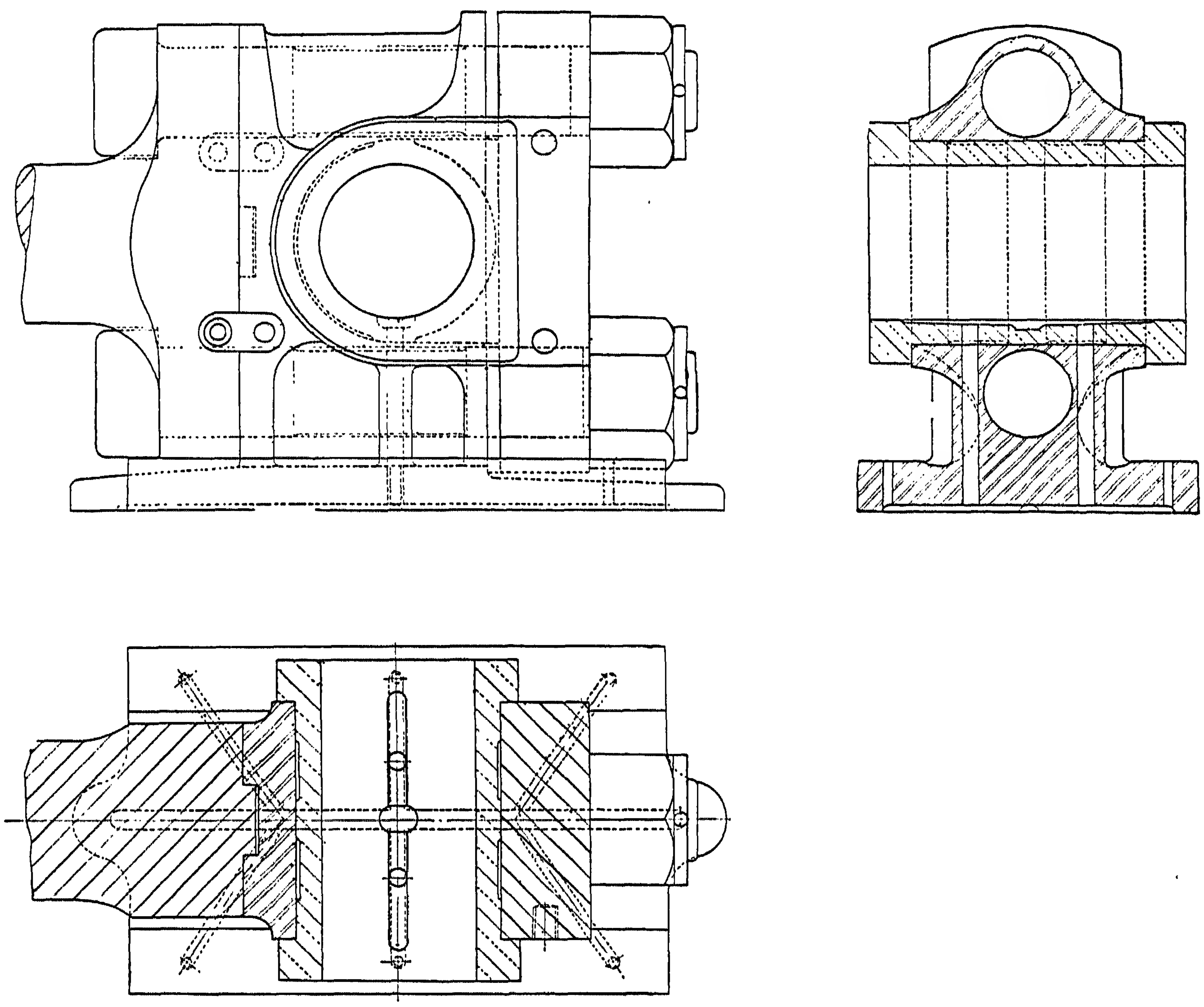


Fig. 41.—Howden Crosshead

The type with fixed gudgeons is shown in fig. 42. It will be seen that these form part of the main block, usually a steel forging to which the piston-rod is secured by a taper and a parallel part, as in the case of the piston-rod. This design allows the latter to be made from a plain bar. The slippers are made of cast steel, and are secured to the body of the crosshead by either screws or studs. In the case of bored guides the slippers are finished in a grinding machine. Some makers provide collars at the outer end of the gudgeon-pins, and this course seems preferable, as it helps to prevent the jaws of the connecting-rod from spreading outwards due to the load.

The guides are of cast iron and are generally of the pendant bored type introduced by Messrs. Belliss & Morcom. In most cases they are cast

with the distance piece. Other makers prefer flat guides cast with the frame, which gives a solid backing to take the thrust due to the obliquity of the connecting-rod. A point in their favour is that the working parts are more open for inspection and adjustment, and when the doors are off, such parts are as accessible as in an open engine. In the case of the bored variety, openings are, of course, made at the sides to give accessibility.

Connecting-rod.—In high-speed engines the ratio of length of connecting-rod to radius of crank is from 5 to 7. The longer the rod the less is the disturbing influence of the inertia forces at the end of the stroke, and the pressure on the guide is reduced owing to the obliquity of the rod being less. At the same time the height of the engine is increased, and this is sometimes an important consideration where head room is restricted. The design of the top end of the rod is determined by the type of crosshead used. In that type in which the crosshead pin is fixed usually by shrinkage in the jaws of the rod, the width of the jaws may be $0{\cdot}6d + 0{\cdot}2$, and the thickness of metal round the eye $\frac{d}{3} + \frac{1}{8}$ in., where d is the diameter of the pin. The depth of the jaw must be sufficient to clear the nuts on the crosshead bolts. The sides of the jaws are usually made a straight taper. The diameter of the rod at the junction with the jaws may be a little less than the diameter of the piston-rod, and the body of the rod increases in diameter by a straight taper to the foot.

In the other design in which the gudgeon-pins form part of the crosshead, the bearings are carried by flat palms formed on the fork of the rod. There are thus two bearings, the length of each being about the same as or 25 per cent more than the diameter, and the total area of these bearings must be such as to give a pressure of from 1000 to 1200 lb. per square inch. This figure also applies to the design first described, in which the crosshead pin is fixed in the jaws of the connecting-rod. The total area of the bolts at the bottom of the thread may be such as to give a stress of 3500 to 4000 lb. per square inch. The width of the caps and of the palms forming the rod may be about 0·7 of the length of the bearings, and their thickness may be the same as or a little greater than the diameter of the bolts. The stress upon them due to bending should be checked, and should not exceed 8000 lb. per square inch. The bending moment should be taken as $\frac{PL}{8}$, where L is distance between bolts and P the total pressure on high-pressure piston. The steps as in the case above described, that is, when incorporated with the crosshead, should be of phosphor bronze or hard gun-metal. The thickness of metal in the sides of the fork should be such that the stress due to the piston load should not exceed 9000 lb. per square inch, when calculated upon a basis of combined direct tension and bending. This should be checked at several sections.

The bottom end is invariably of the marine type. In smaller engines the steps are made of gun-metal lined with white metal, but in medium-size and larger engines the steps are made of cast steel also lined with white

metal. This is cheaper, and enables the weight to be somewhat reduced. The bore is, of course, fixed by the diameter of the crank-shaft, and the length of the crank-pin is such as to limit the bearing pressure to 450 lb. per square inch. The width of the steel cap and of the palm of the rod may be about 0·7 of the length of the crank-pin, and the thickness of the cap may be equal to the diameter of the bolts, or such as to give a bending stress of 8000 to 9000 lb. per square inch for mild steel, and 6500 to 7000 for cast steel, when the bending moment is taken as $\frac{PL}{8}$, P being the total pressure on high-pressure piston, and L is the centres of the bolts. The stress on the bolts may be taken as from 5000 to 6000 lb. per square inch. Double nuts are used for all connecting-rod bolts with the addition of a split pin. The top has a projection fitting into a groove or circular recess in the foot of the connecting-rod to keep the whole structure in position and to resist to some extent the inertia forces caused by the swinging of the rod when working. Circumferential grooves are machined in the white metal to distribute the oil entering from the crank-pin. These grooves are off the centre in each half, in order to prevent a ridge being worn on the crank-pin.

Eccentrics.—The eccentric pulleys are often made of cast iron with the smaller part of steel, but some makers prefer to make the whole of malleable cast iron. The smaller part is usually provided with a tongue fitting into a corresponding recess in the other part, in order to locate the two parts. When of steel it may have a thickness of $\frac{d}{8} + \frac{1}{4}$, where d is the diameter of the shaft.

The two parts of the eccentric are usually held together by cotter studs screwed into the smaller part. The diameter of these studs is made as large as can be got in, and the cotters may have a breadth equal to the diameter, and a thickness of a quarter of that amount.

The width of the eccentric may be about half the length of the crank-pin. There is usually so much surface that the pressure is satisfactorily low. It should not exceed 100 lb. per square inch, when calculated on the basis used for the design of valve rods.

The eccentric straps, whether of cast or malleable-cast iron, should be lined with white metal.

The two studs for securing the foot of the eccentric rod may have the same diameter as the valve-rod bolts, and the bolts holding the two parts together may be a little thicker. The key for the eccentric may have a breadth of $\frac{d}{6} + \frac{1}{8}$ in., where d in. is the diameter of the shaft. The thickness may be half the breadth.

Valve-rod and Eccentric-rod.—The proportions of the valve gear are usually based upon the forces set up by the inertia of the heaviest valve in the engine, usually, of course, the valve of the low-pressure cylinder. As the ratio of the radius of the eccentric pulley to the length of the eccentric rod

is always very great, the force may be taken as equal to $\frac{Wv^2}{gr}$, where W lb. is the weight of the valve or valves if two are used, v ft. per second is the velocity of the eccentric circle, and r ft. is the radius. This gives the total force due to the inertia of the valve. In addition, there is the friction of the valve itself and of the valve-rod packing, and, as these are indefinite quantities, it is usual to take twice the force found above and then proceed with regard to the stresses at the bottom of the thread in the valve rod and also the bolts, as in the case of connecting-rods. The area of the valve-rod pin may be decided in the same way, but the bearing pressure is usually kept lower than that on the crosshead pin or gudgeon, say 500 to 600 lb. per square inch. Generally the design of these parts is the result of experience and is greatly decided by their "look" and mutual proportion to each other

Crank-shafts.—It is almost impossible to compute the actual stresses on crank-shafts of multi-cylinder engines due to the combined effect of twisting and bending. The varying forces and bearing reactions are much too complex. In practice, formulæ are applied which experience has shown to give good results.

A formula often used for the diameter of the crank-shafts of high-speed steam and similar engines is of the form $d^3 = C \times \frac{HP}{N}$, where C is a constant, but which varies for different types of engine. Now $\frac{HP}{N}$ is proportional to a torque, for, taking $HP = \frac{2\pi FRN}{33000}$, where F lb. is a force applied at radius R ft., and N is r.p.m., $\frac{HP}{N} = \frac{2\pi FR}{33000}$ and is thus proportional to the torque FR. The formula given above is seen to be of the right form, for the resistance of a circular shaft to both bending and twisting varies as the cube of the diameter. $F \times R$ is, of course, the mean torque due to the high pressure developed, but, owing to the variations in the turning effort on the crank-pin, the maximum torque is greater. In compound engines it is 1·8 times the mean, and in triples it is 1·4 times.

The constant C, however, allows for all these considerations, and varies from 150 to 170 for compound engines with opposite cranks to 130 to 150 for triple-expansion engines with three cranks placed at 120° to each other.

It is usual to make the crank-shaft of uniform diameter throughout, with the exception that the bearing nearest the fly-wheel is made about 10 per cent larger in diameter, to allow for the stress induced by the over-hanging fly-wheel and that portion of the weight of the armature or other heavy part which may be taken by the outer engine bearing. For the same reason the outer bearing is increased in length also.

The crank-pins are usually of the same diameter as the body of the shaft, and the length is such as to keep the pressure due to the piston load not more than 450 lb. per square inch.

The crank cheeks or webs usually have a depth equal to 1·1 or 1·15 times the shaft diameter, and the breadth may be 0·6 to 0·65 times the diameter. The web nearest the fly-wheel bearing is usually a little thicker, say 0·7 to 0·75 of the diameter.

In the case of triple-expansion engines the crank webs for the high-pressure and intermediate-pressure cranks may be made thinner than above, as the shaft is much stronger than is necessary, especially at the high-pressure end (assuming, of course, that the power is given off at the low-pressure end), and the high-pressure and intermediate-pressure webs may have a breadth of 0·55 of the shaft diameter. Holes for lubrication are drilled through the shaft, as shown in fig. 42. This method was introduced by Messrs. Belliss & Morcom.

Governor.—The governor used with high-speed engines is nearly always mounted on a spindle fixed directly to the crank-shaft, or in some cases to an extension of the shaft, at the high-pressure end. As it acts in a horizontal position, it is, of course, spring controlled. This type is now sometimes called a "crank-shaft" governor, but formerly this term was reserved solely for governors which controlled the position of the eccentric, the regulation of speed and power thus being accomplished by an alteration to the cut-off. Weights controlled by centrifugal force and springs were used, and many attempts were made to improve the action of this type by disposing the weights in such a way that their inertia could come into play to supplement the centrifugal force, but all these types have been abandoned as being quite unsatisfactory. A large number of parts including links and joints were introduced, which rapidly wore, causing considerable lag between an alteration in the load and the adjustment of the speed.

The type of governor now universally used is mounted on a spindle forming an extension of the crank-shaft and controls the throttle valve, but in some cases the high-pressure piston valve is given a partial rotation at the same time through a relay cylinder actuated by the oil pressure system or by mechanical means, also under the control of the governor. The steam edge of the high-pressure piston valve is serrated, one side of the serration being parallel to the axis and the other edge being at an angle, the valve liners having ports with similar edges. The effect of rotating the valve is to increase or decrease the lap, and therefore to shorten or lengthen the period of steam admission. The lead is modified at the same time, so that with early cut-offs there may be no lead at all or even negative lead, which means, of course that when the piston is about to commence its stroke, the valve has not then opened to steam. With this gear wide regulation is not possible by alteration to the cut-off alone. It is generally limited to the range between three-quarter load and overload, the throttle valve taking charge of the lower loads.

The governor is enclosed in a casing fixed to the end of the bedplate and crank-case, and all the parts are lubricated by oil under pressure, taken from the lubricating system.

An adjusting spring attached to a fixed part of the crank-case at one end

and to a lever on the governor rocking arm at the other, with a hand wheel for regulating its pull, is always provided, and is of such a strength that it gives a total variation of 10 per cent in the engine speed, that is, 5 per cent above and below the normal.

There have been many types of throttle valve, but that introduced by Messrs. Belliss & Morcom in the early days of their engine has practically superseded all others on account of its simplicity. As will be seen from fig. 42, it consists of a casing in which is suspended the double valve seat. The valve itself has no definite closing seat, and is perfectly balanced. The seat and valve are made of gun-metal in the smaller sizes, but cast iron may be used for the larger sizes.

The spindle is attached by a nut and collar to the valve, and passes through a long sleeve in the cover, the sleeve being usually made of gun-metal. As the valve requires little force to move it, the spindle may be of quite small diameter, and this is an advantage in helping to reduce the amount of leakage through the sleeve. A cup fixed on the rod from the governor receives the leakage, whence it may be drained by a small pipe to the high-pressure distance piece or other convenient place.

There is, of course, a force acting upon the end of the spindle equal to the area of the cross section multiplied by the steam pressure of the moment on the cylinder side of the valve, and this force should be taken into account when computing the strength of the springs. With this type of governor and throttle valve it is quite possible to obtain steady governing with 2 per cent variation between full load and no load, but this is too close for ordinary purposes, a variation of 5 per cent being usual.

When the two methods of governing are compared with respect to their effects upon the total steam consumption, for the range considered, it is found that the result depends to a certain extent on the load. There are two loads for which "throttle" governing and "cut-off" governing give identical results. For loads between those two, the consumption is in favour of cut-off governing, but at lower and higher loads "throttle" governing has a decided advantage.

If it happens that the normal full load of the engine is not far from the load corresponding to one of the neutral points just mentioned, and if the nature of the duty is such that the engine will be running at nearly that load for most of the time, then, obviously, no advantage is given by the additional complication of cut-off governing; but if the load on the engine fluctuates considerably below full load, then there will be an advantage in its adoption.

On the other hand, if the engine is likely to run at light loads for any considerable portion of the time, then throttle governing would give some advantage.

It is a mistake to have the throttle valve too big. The governing is then done through a smaller range of movement, and therefore of speed, and hunting at light loads is likely to occur.

The cut-off gear is also useful for obtaining occasional overloads, being

much more economical than the arrangement sometimes used to by-pass high-pressure steam to the intermediate valve chest in the case of triple-expansion engines, or to the low-pressure valve chests in the case of compound engines.

The weights consist of bell-crank levers which are pivoted in a casting firmly fixed to the governor spindle, thus rotating with it. The short end of the bell-crank lever engages with a sleeve which slides upon the spindle and rotates with it. Keys are not used, owing to the friction that would be caused. Stops are cast on the sleeve, upon which the weights rest under the tension of the springs when the engine is not running. At the end of the sleeve a circumferential groove is formed in which a split collar, usually of gun-metal, works, each half of the collar having a projection upon it which fits into a hole in the end of a forked bell crank lever, or there may be two levers. The spindle upon which the lever is mounted may be fixed in position by the centre-point bearings to reduce friction, the centre points being carried in a bracket, supported from the governor case. From the other end of the bell-crank lever a rod is led to the throttle valve, arranged if possible vertically above it, a direct connection being preferable in order to avoid a multiplicity of joints. Plain pin joints are used for the connections, and the surfaces should be ample, as although the force required to move the throttle valve is small yet there is always a certain amount of vibration and movement in the governor, causing wear and ultimately leading to lost motion.

There are invariably two springs, one on each side of the governor weights, attached at each end to hooks or pins fixed in the weights, so that each spring takes half the centrifugal force. When making calculations it must be noted that the force generated in each weight forms the reaction to the force generated in the other weight, so that the springs may be calculated for the varying force in one weight only, bearing in mind at the same time that the total extension of the spring for that force is double that given by the radial motion of one weight. It is as though each spring were cut in the middle, and each half-spring fixed to some part rotating with the governor, but that in manufacture the half-springs had been joined together to make one spring.

If it be imagined that the centre of gravity of the balls could be made to coincide with the axis, and that the springs were arranged in such a way that their pull on the balls in that position were zero, the combination in that form would be useless. With a given weight and strength of spring there would be a certain definite speed or angular velocity at which the balls could take any position, for the centrifugal force would vary with the radius, and so would the tension of the spring, and the slightest variation in speed in either direction would cause the governor to move to one or other extreme position. It would be impracticably sensitive. For stable governing it is necessary that a certain variation of speed be fixed upon, and that the strength or pull of the springs should increase more rapidly than the centrifugal force of the balls as they move outwards, *assuming the*

speed to remain constant. For the weights to move outward, it would then be necessary for the speed of revolution to increase, and there would be a definite position of the balls corresponding to each speed throughout the range. The governor would then be stable.

Fig. 43 shows these relations. The radius of the centre of gravity of the weights is set down on the horizontal axis, and the centripetal or inward controlling forces, exerted by the springs upon the weights, are marked on the vertical axis.

Let us assume that the line OF_c coincides with the axis of the governor, and that the lines A radiating from the origin O show on the same scale as F_c the centrifugal forces generated in the balls at different speeds and varying radii. Each of these inclined lines corresponds to a definite speed, and

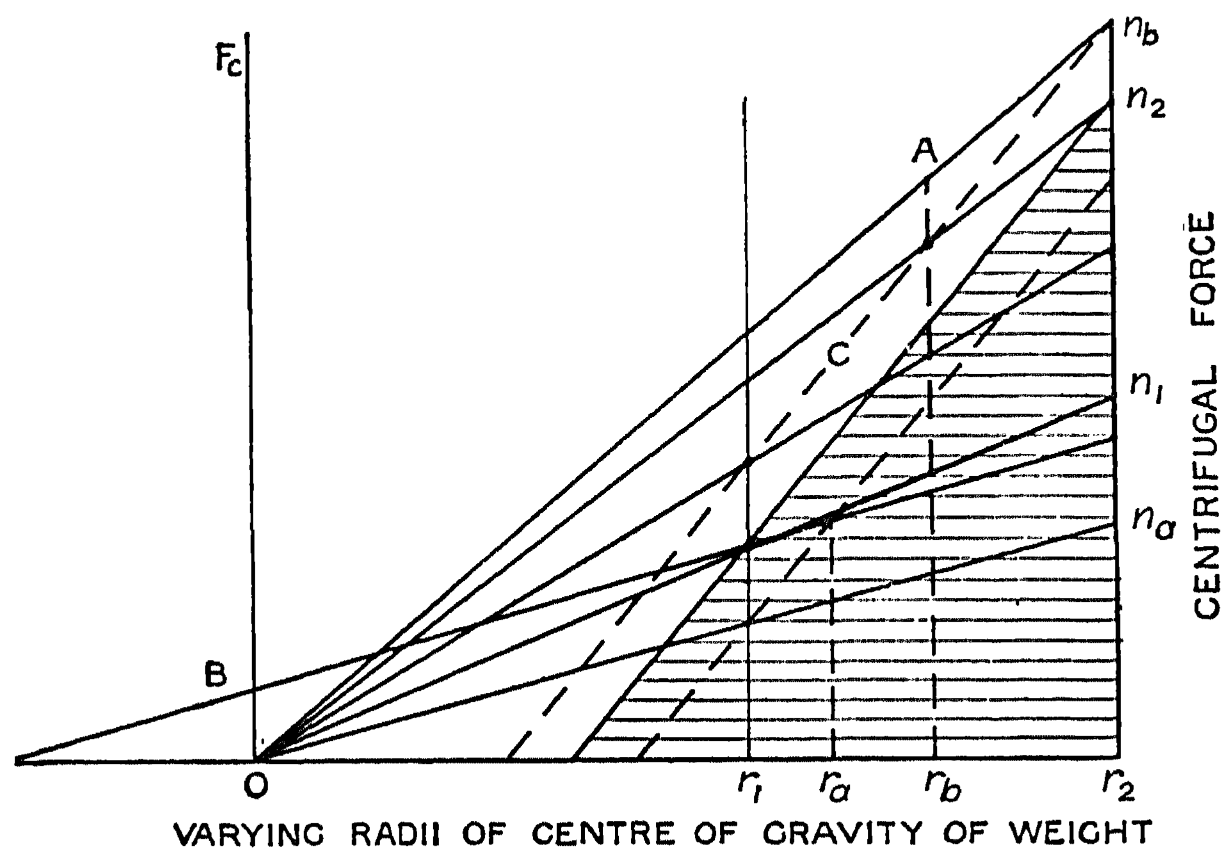


Fig. 43

the higher the speed the greater their slope. The centrifugal forces vary, of course, directly with the radius for any given speed of revolution. If the springs were so fixed that at the point O—that is, at the centre of the governor spindle—their pull was zero, then the balls could take up any position on any given line so long as the speed remained constant, and the governor would be in a condition of neutral equilibrium, or isochronous. Assume the engine to be standing, and the balls under the tension of the springs resting upon a stop with their centre of gravity at radius r_1. On the engine being started, the balls would remain on the stop until the centrifugal force was equal to the pull of the springs in that position, but immediately the speed increased, no matter by how little, the balls would move outwards throughout their range, cutting off the steam completely. The speed of the engine would then decrease, and the balls would commence to return to their inner position and full steam would be given to the engine. Or the balls might rest somewhere between their extreme positions for a little while, only to move away by an indefinite amount as soon as the load varied and the speed of the engine altered.

Assume now that other springs were substituted, having their position

of zero pull on the side of the axis opposite to the weights. The action of these springs is shown by the line B on the diagram. The governor balls are resting on the stops as before, and would stay there until the speed became high enough for the centrifugal force to overcome the pull of the springs in that position, when they would immediately fly out to their limit with some violence, and this action would be reversed immediately the speed fell. Such a governor would be in unstable equilibrium. The line B, as it progresses in the direction of r_1, cuts lines A of centrifugal force corresponding to lower and lower speeds of revolution. It is less steep than the lines A.

It is evident that a governor with either arrangement of spring or controlling force described above would be useless.

If we now arrange the springs in such a way that their zero pull on the ball is when the centres of gravity of the latter are at some distance from the axis, and their pull, represented by the line C, is steeper than the lines n_1 and n_2, representing the limits of speed between which the engine is designed to run, then the governor would be stable, for in order to stretch the spring the balls must move out radially, and this can be done only by the speed increasing and vice versa. There is thus a definite position for the balls for any given speed.

The area between that part of the line C, which intersects lines n_1 and n_2, the ordinates at r_1 and r_2, and the base, is a work diagram representing the amount of energy stored in the governor through its range of action.

The effect of friction in the governor and the parts connected to it have to be considered.

If we assume this to be constant in amount and opposing motion in both directions, we may show its effect by drawing on the diagram two lines parallel to the line C, one above and the other below, at a vertical distance, which, when measured on the scale F_c, is equal to the frictional resistance as indicated by the two dotted lines parallel to C. We shall notice from the diagram that the line of centrifugal forces n_2 is in equilibrium with the upper friction line at a radius r_b less than r_2, and n_1 is in equilibrium with the lower friction line at the radius r_a greater than r_1.

The effect is, that in order that the balls may move out to t_2 and move in to t_1, the speed must be increased to n_b and decreased to n_a respectively, the total speed variation then being $n_b - n_a$ instead of $n_2 - n_1$. The power available is also decreased.

Fly-wheels.—The turning moment varies continuously throughout the revolution of an engine. In a single-crank engine and a double-crank engine with the cranks opposite to each other, there are two maxima and two minima positions, and in a three-crank engine with the cranks disposed at equal angles there are three maxima and minima positions. The mean effort, of course, lies between these extremes, and it is the amount of excess work generated during a maximum period, together with the defect from the mean effort during a minimum period, that causes a certain fluctuation of speed. It is the function of the fly-wheel to make this irregularity as

small as possible by absorbing kinetic energy during a maximum period, and giving it out during the minimum period which immediately follows.

With fly-wheels of practicable size, some cyclical speed variation, of course, is inevitable, and the range depends upon the ratio of the total work stored in the fly-wheel, in the form of kinetic energy, to the work taken in and given out during the greatest maximum or minimum periods respectively. The algebraical sum of the energies taken in and given out is, of course, zero. At the end of a minimum period the speed is at its lowest. It is then accelerated and passes through the mean speed to the highest, and goes through these changes in the reverse order during the ensuing minimum period. The ratio of the variation to the mean speed, or $\frac{V_2 - V_1}{V_0}$, is called the coefficient of fluctuation of speed, and is usually denoted by $\frac{1}{M}$. This varies from $\frac{1}{200}$ to $\frac{1}{300}$ for electrical generating purposes.

It is computed as follows when the weight of the fly-wheel rim is known.

Let W lb. denote the weight of rim, V_2 and V_1 the higher and lower speeds respectively, and V_0 the mean speed in feet per second. Let the excess energy, as obtained from the turning moment diagram, be

$$\Delta E = \frac{(V_2^2 - V_1^2)W}{2g};$$

$$\text{but} \quad V_2^2 - V_1^2 = (V_2 + V_1)(V_2 - V_1),$$

$$\text{and} \quad V_2 + V_1 = 2\,V_0;$$

$$\therefore \quad \Delta E = \frac{(V_2 - V_1)\,V_0 W}{g}$$

$$\text{and} \quad \frac{V_2 - V_1}{V_0} = \frac{g\Delta E}{WV_0^2}.$$

$$\therefore \quad \frac{1}{M} = \frac{\Delta E}{2E},$$

$$\text{when } E = \frac{WV_0^2}{2g},$$

the energy in the rim at mean speed.

The weight of the rim is calculated from $W = g.\Delta E.M/V_0^2$. The energy E in the rim must be computed by taking its weight to act at the radius of gyration $r = \sqrt{\frac{r_1^2 + r_2^2}{2}}$. The energy contained in the web is usually neglected in this calculation, so that there is no great error in taking r as the mean of the external and internal radii.

In compound engines with two cranks the ratio of the excess work to the work done during one revolution is 14 per cent to 15 per cent and in triple-expansion engines it is 8 per cent to 9 per cent. It would seem, then, that in the case of the latter, the fly-wheels might be comparatively lighter; but one of the functions of a fly-wheel is to keep within limits the momentary rise in speed which would occur should the

whole load suddenly be taken off. The speed would then rise, causing the steam supply to be reduced by the governor, but the speed of the engine would be still accelerated by the expansion of the steam left in the cylinder and passages. In the case of a triple-expansion engine the amount of steam left in the ports and passages is usually greater than in a compound engine of the same speed and power. Of course the momentary rise in speed caused by a sudden decrease at lower loads is proportionately less, with a given engine, because the pressure of the steam in the cylinder and passages of the engine is lower, and therefore can do less work in acceleration.

For these reasons the proportions of fly-wheels are greatly empirical. For compound engines the stored energy in foot tons is from 0·3 to 0·5 per brake horse-power and for triple engines it is from 0·45 to 0·75. In each case it may be less if the machinery immediately driven by the engine has a rotor containing a considerable amount of stored energy.

Fly-wheels for high-speed engines are of a very simple type. They are invariably a plain casting consisting of a heavy rim connected to the boss by a web. For heavy wheels the boss is sometimes made separate, and is keyed on to the shaft, the wheel being mounted upon the boss. This construction is likely to minimize casting strains in the web, the central hole allowing it to contract in cooling much more easily. It is also of some advantage in transport, as the wheel can be dispatched separately from the shaft. There is a flange on the boss against which the machined face of the web abuts, the whole, with the bolts, forming a coupling for connection to the coupling on the driven shaft or the armature of the electrical generator.

The peripheral speed of the rim should not exceed 100 ft. per second. In small engines it is often not more than 80 ft. per second.

The coupling bolts are usually in single shear, and their diameter should be such as to keep the shearing stress about 6000 lb. per square inch. If R is the radius in feet to the centre line of the bolts, then the shearing force acting at $R = \text{BHP} \times \dfrac{33{,}000}{2\pi RN} \times 1{\cdot}8$ for compound engines and by 1·4 for triples. N is the number of revolutions per minute.

The coupling boss on the shaft has a diameter of twice that of the shaft, and the thickness of flange may be equal to $1\frac{1}{4}d + \frac{1}{4}$ for small bolts and $1\frac{1}{4}d + \frac{1}{2}$ for large bolts where $d =$ diameter of bolt. The bolt holes should be left for reamering, and the bolts made a light driving fit.

STEAM TURBINES

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Steam Turbines

CHAPTER I

Introduction

The development of the modern steam-turbine industry dates from the almost simultaneous inventions of the Hon. Sir Charles Algernon Parsons and the late Dr. Carl Gustav Patrick de Laval in the early eighties of last century. The progress of this industry both as regards design and application was comparatively slow up to the beginning of the present century, but since then it has made extraordinarily rapid strides, with the result that to-day the steam turbine ranks as the foremost prime mover for all large power units.

The conversion of heat energy into mechanical energy is effected in a steam turbine by utilizing the kinetic energy acquired by steam in expanding from a higher to a lower pressure.

The problem confronting the designer is to ensure that the desired expansion approaches the adiabatic line as closely as possible, and that the kinetic energy attained is converted into mechanical energy in the most efficient manner.

The external losses, namely those due to bearing friction, &c., and to radiation, are relatively small and unimportant in a steam turbine. The conditions of efficient expansion are so well understood to-day that the losses in this direction are also generally reduced to a minimum, but there is still a margin for improvement in the methods of utilizing the kinetic energy, and it is particularly due to variations in this direction that individual turbines differ in regard to their over-all efficiency.

Types.—Broadly speaking, turbines are divided into two main types, namely, the impulse and reaction types.

Impulse Turbines.—Where the expansion takes place in fixed blades or nozzles, and the energy is transferred to revolving blades without further pressure drop, the turbine is known as an impulse turbine. In the De Laval turbine, which was the first practical impulse turbine, the steam was expanded over its full range of pressure in a single nozzle or batch of nozzles, and the energy was utilized in a single row of blades mounted on a wheel or disc.

Even with a comparatively small range of expansion the steam velocity so attained becomes exceedingly high, and calls for a very high speed of revolution for its efficient utilization, and necessitates the use of speed-reduction gearing for most purposes. Such turbines are known as single-stage turbines. With a view to overcoming the limitations imposed by such high velocities, Dr. Zoelly, of Zurich, and Professor Rateau, of Paris, applied the principle of subdividing the total available pressure drop, and built multi-pressure stage turbines. These consist of a number of discs similar to that used by De Laval, with blades fixed on the periphery. The discs are mounted on a common shaft with diaphragms carrying the nozzles or guide blades placed between. Other inventors retained the single-pressure stage, but mounted two or more rows of blades on the disc with intermediate guide blades which guide the steam from one row of blades to the next without fall of pressure, thus subdividing the steam path into velocity stages. Such turbines have the advantage of retaining the single wheel, but the efficiency attained does not equal that of a multi-pressure stage turbine.

The original turbine due to Curtis, an American inventor, combined the two methods of subdivision, that is to say, it consisted of a number of pressure stages, each of which in turn consisted of two or more velocity stages.

Some of the largest turbines are still being built on the single multi-pressure stage principle, but the majority of designs of impulse turbines in all countries now take the form of a two-row velocity stage for the first-pressure stage, and simple-pressure stages for all lower stages.

The introduction of the "velocity compounded wheel" in the first stage means a slight fall in the efficiency, but has the twofold advantage of reducing the number of discs and at the same time allowing a greater drop of pressure in the first batch of nozzles with resultant lower steam pressures and temperatures in the turbine housing proper.

Reaction Turbines.—If the expansion of the steam is effected in the moving blades, the mechanical energy is derived from the maintained reaction due to such expansion, and such turbines are known as reaction turbines. This term is generally applied to all turbines embodying the reaction principle in all stages, regardless as to whether or not the turbine contains fixed blades in which the steam is partially expanded before entering the revolving blades. By far the most important type of reaction turbine is that due to Parsons, in which the steam is expanded in both fixed and moving blades. The only practical turbine in which the whole range of expansion is effected in revolving blades is that designed by Messrs. Ljungström. The first turbines built by Parsons, and all subsequent designs built on the reaction principle, embodied the multi-pressure stage principle, and it is no doubt due to this feature of the inventor's original designs that they found a more rapid and more extensive application than those built by his contemporary Dr. De Laval, and it is important to note in estimating the value of his invention that the original principles of his turbines have been retained to this day.

The rotor construction of Parsons turbines differs essentially from that

adopted for most impulse turbines. It has already been explained that the moving blades of impulse turbines are generally mounted on the periphery of separate discs which are threaded on to the shaft. If the same method of construction were adopted in a reaction turbine, the drop in pressure across the individual discs would cause a heavy and varying axial bending load on the discs which would be very undesirable, and this applies particularly at the high-pressure end. It is therefore customary to enlarge the diameter of the shaft within the turbine housing to approximately that required for the roots of the blades, and to mount the blades in grooves cut into the periphery of the shaft.

The diaphragms used in impulse turbines are replaced in reaction turbines by a series of blade rings fixed in grooves cut into the body of the casing.

Impulse-reaction Turbines.—Where one or more stages of a turbine are built on the impulse principle and the remainder on the reaction principle they are known as impulse-reaction turbines. In practice such turbines are built with one impulse stage consisting of a nozzle plate and velocity wheel at the high-pressure end, the intermediate- and low-pressure stages being of the Parsons reaction type.

General Description.—Some early turbine designs, notably the earlier Curtis turbines, were built with a vertical axis, but practically all modern turbines are built with horizontal axes.

Owing to the increasing specific volume of the steam with falling pressure, the blade and nozzle areas and consequently the blade lengths increase with succeeding stages.

The Ljungström turbine is the most successful example of the radial-flow type, that is to say, the steam enters around the centre of the turbine and passes radially through expansion stages to the periphery. The majority of turbines are of the parallel-flow type, in which the steam paths are substantially parallel to the axis.

Most steam turbines consist in the main of a casing or body, split horizontally, and a rotor or shaft together with the necessary bearings, bedplate, governor gear, oil-pump, and a thrust block to prevent axial movement. In reaction turbines the unbalanced axial pressure across the areas of the blades and shaft exposed to the steam calls for the use of dummy pistons or balance pistons to counteract the resultant end thrust.

In impulse turbines the unbalanced axial forces are sufficiently small to be dealt with by a thrust block of ordinary pattern.

In some of the latest turbines of the reaction type, the balance pistons have been avoided by the use of thrust blocks of the Michell type, which are capable of handling the very heavy thrust without assuming unreasonable proportions or introducing severe friction losses.

In reaction turbines the fixed blades are let into the body of the casing. In impulse machines the first batch of nozzles is usually formed into one or two ring segments, bolted to the inside of the casing at the high-pressure end, which is then formed into a circular steam passage resembling the letter C in section, with the nozzle plate bolted across the gap.

Alternatively the nozzle plate may be bolted to a separate nozzle box, which in turn is bolted against an opening at the end of the top half of the casing.

The fixed blades for the remaining stages are cast into split diaphragms, which carry distance rings at their periphery so as to provide the necessary spacings for the revolving discs let into the casings and butting against each other.

The diaphragms are usually dished to give the required strength, and are formed with openings in the centre through which the shaft passes, the inside periphery being provided with baffles which give a running clearance for the shaft and prevent excessive leakage losses across the annular clearance spaces between the diaphragms and the shaft. In all but exceptional designs the diaphragms are made in two halves, with a tongue and groove joint along the horizontal plane. The upper halves of the diaphragms are fastened to the top half of the casing, so that when this part has been lifted the rotor can be examined or taken out as a whole.

Casing.—The casing is generally made of cast iron, but where high superheats are used it is desirable to split the casing vertically and make the high-pressure end of cast steel, or, where the nozzle-box construction is adopted, to make the latter of cast steel.

Couplings.—Continental designers largely prefer the use of solid couplings between the turbine and the driven machine, but flexible couplings are usually adopted in this country except on small units.

The flexibility of such couplings is limited, but suffices to take up the movement of the shaft ends due to their normal deflection. The desirability, amounting in practice to necessity, of enclosing the coupling at turbine speeds, and the consequent impossibility of avoiding the presence of hot-oil vapours, prohibits the use of leather, which is generally considered the best driving medium in a flexible coupling. A large number of designs have been tried from time to time, but the simple steel-claw type and the tooth type have now become more or less standardized.

Heat Expansion.—Provision has to be made in all turbines for taking up the expansion of body and rotor. The large and often rapid variations in temperature which occur with variations of load, and especially when changing from condensing to non-condensing and vice versa, have given rise to great difficulties, and it is probably safe to say that the largest number of breakdowns to turbines have owed their initial cause to unlooked-for movements or distortions of body or rotor due to temperature variations. To minimize this danger the design should embody rigidity at fixed points, and easy sliding surfaces where expansion is intended to take effect. At the same time, symmetry and simplicity of construction of the body casting, and absence as far as possible of sharp changes in diameter or of unnecessary ribs or attachments, are matters calling for close attention. It is, of course, of primary importance that expansion should not affect the alignment of the plant.

Blades, Materials, and Wear.—The blades of reaction turbines,

both fixed and revolving, as well as the distance pieces between blades, are drawn to required section, cut to correct length, the material used being generally brass or bronze. The methods of construction of impulse blading vary to some extent and are described later.

In impulse turbines steel alloys, generally nickel steel, are used as blade material. Latest practice favours the use of "rustless steel", with a view to reducing the risk of corrosion which has proved a serious trouble from time to time in impulse turbines, and the opinion is held that not only the composition of the material but also the degree of smoothness of the blade surfaces and edges has a bearing on the life of the blades.

The trouble can usually be traced to the presence of water or other impurities in the steam. Even when the steam is initially dry or superheated it will be found to be wet in the lower stages. The purity or otherwise of the feed-water is, of course, an important factor.

The problem is not often met with in reaction turbines owing to the lower steam velocities employed, and in fact such turbines have been opened up after more than ten years' running and have shown no sign of wear on the blading.

Governing.—The turbine speed is regulated by a centrifugal governor gear, driven from the main shaft.

All but small-size modern turbines are governed on the relay principle, that is to say, the movement of the governor is transmitted in the first place to an auxiliary oil- or steam-valve system, and the power required for actuating the main-throttle valve, &c., is obtained from an independent source—either an oil-pump or the main steam supply.

In this manner the work to be performed by the governor is reduced to a minimum, and the governor construction can be of the comparatively delicate nature required for sensitive action.

The governor is generally provided with a speed-variation gear, which allows a speed adjustment of about 5 per cent independently of the load. This gear can be hand-operated, and may be arranged for electrical operation by means of a small motor which can be controlled from the switchboard.

The valve system takes one of three forms. First, pure throttle control, in which case the throttle-valve opening is adjusted by the governor through the relay over the full range of load. Second, combined throttle and nozzle control. Here the nozzle chest and plate are divided into a number of compartments, usually three, each of which is controlled by a separate valve and representing half, full, and overload respectively. Below half load the first nozzle valve only is open, and the throttle-valve opening varies from zero to full opening from no-load to half load. At full load the second nozzle valve opens, and the steam is then admitted to the second nozzle batch, the main throttle-valve opening being reduced to give a reduction in pressure corresponding to the increased nozzle area and increasing again to full open at full load, at which stage the third or overload valve comes into operation. By this means a better pressure

distribution and steam consumption is obtained at light loads than with simple throttle control.

Third, pure nozzle control, in which the nozzle plate is divided into a large number of compartments controlled by cam-operated tappet valves which come into operation successively.

It is usual to provide turbines with an emergency governor, in addition to the ordinary speed-regulating governor, for the purpose of assuring that steam is cut off completely in case the main governor should fail to hold the turbine to its normal speed.

The emergency governor should come into operation at a speed of 10 to 12 per cent above normal running speed, and, except in the smallest turbines, it is desirable that valves actuated by this governor should have no other function than to cut off the whole of the steam.

Overloads.—Where steam is admitted to the full blade or nozzle area at the first stage, as in the case of reaction turbines, any overload required has to be obtained by means of a by-pass which admits steam at full pressure to a lower stage of the turbine. This involves a more difficult casing construction, and has the drawback of subjecting a greater portion of the turbine to maximum pressure and temperature.

The overload valves on later turbines are automatically operated, and it is usually possible to obtain an overload of 25 per cent without any substantial fall in efficiency.

It is not now customary to run large steam turbines to atmosphere at all. The condensing plants are made sufficiently reliable to make the complete and sudden failure of the vacuum a very exceptional occurrence. In the case of smaller units it is still common practice to make provision for running to atmosphere, and in such cases it is usual to require the turbine to give an output of about 60 per cent of full load with the aid of the overload valve.

Glands.—Glands require to be fitted at the points at which the shaft enters and leaves the casing to prevent high-pressure steam escaping to atmosphere and air entering the condenser at the lower-pressure end. These glands are usually steam packed, and either consist of a series of carbon rings which bear lightly on the shaft and form a stuffing box, or they take the form of a so-called labyrinth packing. The designs of the latter vary in detail, but are based on the principle of throttling the steam by means of a series of fine annular clearances, formed by corresponding metal rings let into the casing and shaft. The rings are usually made of brass, and are fitted with knife edges to allow for contact occurring without causing damage. In certain designs the glands take the form of a water seal.

CHAPTER II

The Thermodynamics of the Steam Turbine

In a turbine where steam is supplied at a definite pressure and temperature, each pound of steam supplied contains a definite quantity of heat which was obtained by the combustion of fuel in the boilers.

As the steam passes through the turbine it expands, with a corresponding fall of pressure, and on leaving the last row of blading is discharged, usually to a condenser, at some definite lower pressure. During this range of expansion, fixed by the initial temperature and pressure and the final discharge pressure, a definite proportion of the total heat energy of the steam becomes available for conversion into mechanical work, and the degree to which the turbine is capable of converting the available expansion energy into useful work represents the efficiency of the turbine.

Roughly speaking, the principle of the conversion of energy in a turbine may be regarded as occurring in two steps. Firstly, the steam is allowed freely to expand, and in doing so the expansion energy liberated produces kinetic energy in the form of a high steam velocity; then secondly, the kinetic energy of the steam is absorbed by the rotating elements of the turbine, and is thereby converted into useful mechanical work.

A general discussion of the thermodynamic basis upon which the design of heat-engines, and therefore steam turbines, proceeds is given in " Applied Heat ", Vol. IV, p. 123.

The entropy, a quantity which is much used in discussions of the design of turbines, is explained in Vol. IV, p. 206, while the Mollier diagram is dealt with in the same volume, p. 221.

The Thermodynamic Efficiency of a Turbine (η).—The over-all thermodynamic efficiency of a turbine may be defined as the ratio of the useful work delivered at the turbine coupling to the mechanical equivalent of the available heat supplied to the turbine by the steam. When once the value of η is known for a steam turbine, it is possible to calculate the amount of steam required to develop the desired power.

Let H be the heat available from the expansion of 1 lb. of steam in the turbine;
η, the thermodynamic efficiency of the turbine;
W, the weight of steam, in pounds per second, supplied to the turbine, i.e. the water-rate.

Then the useful work, in foot-pounds, done by the turbine, per second (U), is given by

$$U = \eta WHJ,$$

i.e. the horse-power (H.P.) is given by

$$\text{H.P.} = \frac{\eta WHJ}{550},$$

and the steam consumption (C) per hour, per horse-power, is given by

$$C = \frac{3600W}{\text{H.P.}} = \frac{2546}{\eta H}.$$

The phrase " the steam consumption " means the consumption of steam per horse-power, per hour (briefly per horse-power-hour).

Conversely, if the steam consumption C of the turbine be measured, the efficiency is at once given by

$$\eta = \frac{2546}{CH}.$$

In considering more fully the factors which influence the over-all efficiency of a turbine, it is necessary to examine the various inherent losses of energy which occur.

Internal Losses due to Steam Friction.—During the passage of the steam through the nozzle and blade passages which constitute the working elements of a turbine, frictional losses necessarily occur, and in all cases (excepting machines of the smallest output) these constitute the main losses in a steam turbine.

A secondary loss also occurs due to steam friction on the moving surfaces of the discs or rotors.

Internal Leakage Losses.—In the multi-stage turbine a certain proportion of the steam escapes from stage to stage without performing useful work. In the case of the impulse turbine this leakage mainly occurs at the glands where the shaft passes through the diaphragms. In the reaction turbine the leakage space occurs at the tips of the fixed and moving blades.

External Leakage Losses.—At the main glands where the rotor shaft passes through the main casing of the turbine a small leakage of steam occurs. At the high-pressure end of the turbine a certain amount of steam leaks outwards, and at the exhaust end, where the internal pressure is below atmospheric pressure, a quantity of live steam must be supplied to pack the gland and so prevent air being sucked into the condenser.

Radiation Losses.—Owing to the high temperature of the turbine casing, a certain amount of heat is lost to the surrounding atmosphere by radiation, but in a modern turbine of comparatively large output, if the casing be well covered with some form of insulating material, the radiation loss is almost negligible.

External Mechanical Losses.—From the gross horse-power generated by the turbine the frictional losses which occur in the bearings and thrust block must be deducted. Also a small amount of power is absorbed by the oil-pump which supplies the bearings, &c., and by the driving of the necessary governing mechanism. The sum of all the external mechanical losses in the usual design of turbines may represent from 1 to 3 per cent according to the size of machine, and this indicates the difference between the gross horse-power developed by the turbine and the net power delivered at the coupling.

Internal Losses indicated on Entropy Diagram, and Reheat Factor.—When any form of steam frictional loss occurs within a turbine, the energy dissipated by friction is returned to the steam in the form of heat, and this addition of frictional heat produces an increase of entropy.

Similarly, when any interstage leakage loss occurs, the velocity generated through the leakage area is dissipated and returns to the steam as heat.

Referring to the temperature entropy diagram (fig. 1), CD represents the adiabatic expansion line; but if, at each stage of the machine, an increase of entropy occurs due to internal losses, the expansion line in an actual turbine becomes CK where the total internal losses are given by the area FCKL. The complete working cycle of the turbine is thus represented by the area ABCK, and the additional area CKD shows the extra heat available due to reheating of the steam. The ratio of this additional heat to the available adiabatic heat is known as the reheat factor of the turbine, and it usually varies in value from 4 to 6 per cent, according to the steam conditions and the internal efficiency of the turbine.

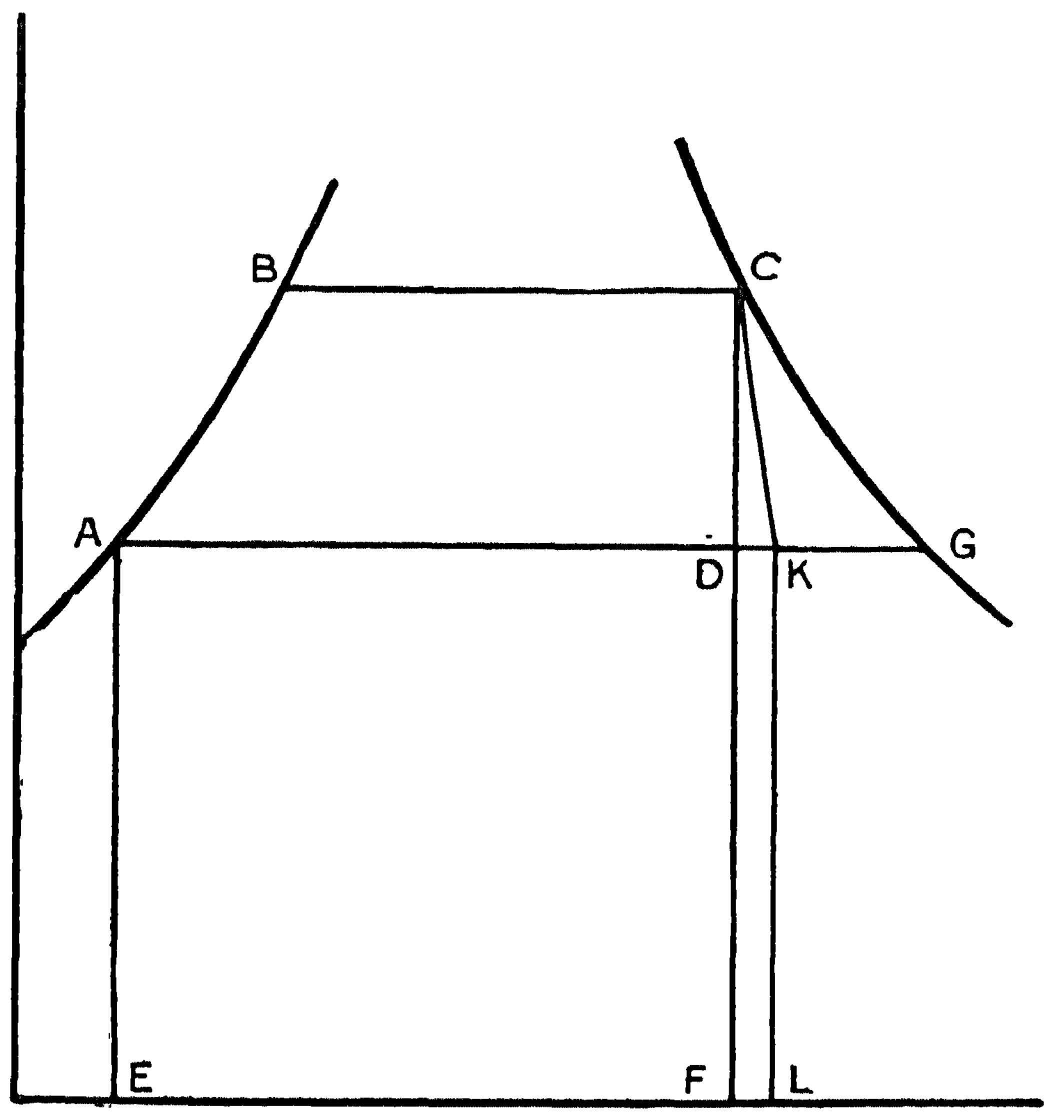


Fig. 1.—Reheat Factor on TQ Diagram

Internal Efficiency of Turbine shown on Mollier Chart.—The expansion of steam throughout a turbine can be clearly indicated on a Mollier chart. In all turbines of moderate output the expansion of the steam is divided into several stages, to each of which is relegated a definite heat drop. Thus, as example in fig. 2, the heat drop in the first stage is indicated by the line AB, the steam expanding from 200 lb. to 55 lb. If now all the losses of energy which occur in this stage be added together and scaled off as the length BD, the point D will represent the total heat contents at the end of the first stage; and projecting this point D horizontally to the pressure line of 55 lb., the point A_1 shows the position on the diagram representing the steam condition at the inlet to the second stage, 55 lb. per square inch, 120° superheat.

A similar construction may now be adopted for each succeeding stage until the final exhaust condition E is reached. The total heat available throughout the turbine is thus given by the sum of the quantities ($AB + A_1B_1 +$ &c.)

and this is greater than the straight adiabatic heat drop AC by a percentage representing the reheat factor thus:

If H is the total adiabatic heat drop,
h, the heat drop per stage,
r, the reheat factor, then

$$\Sigma h = (1 + r)\text{H}$$

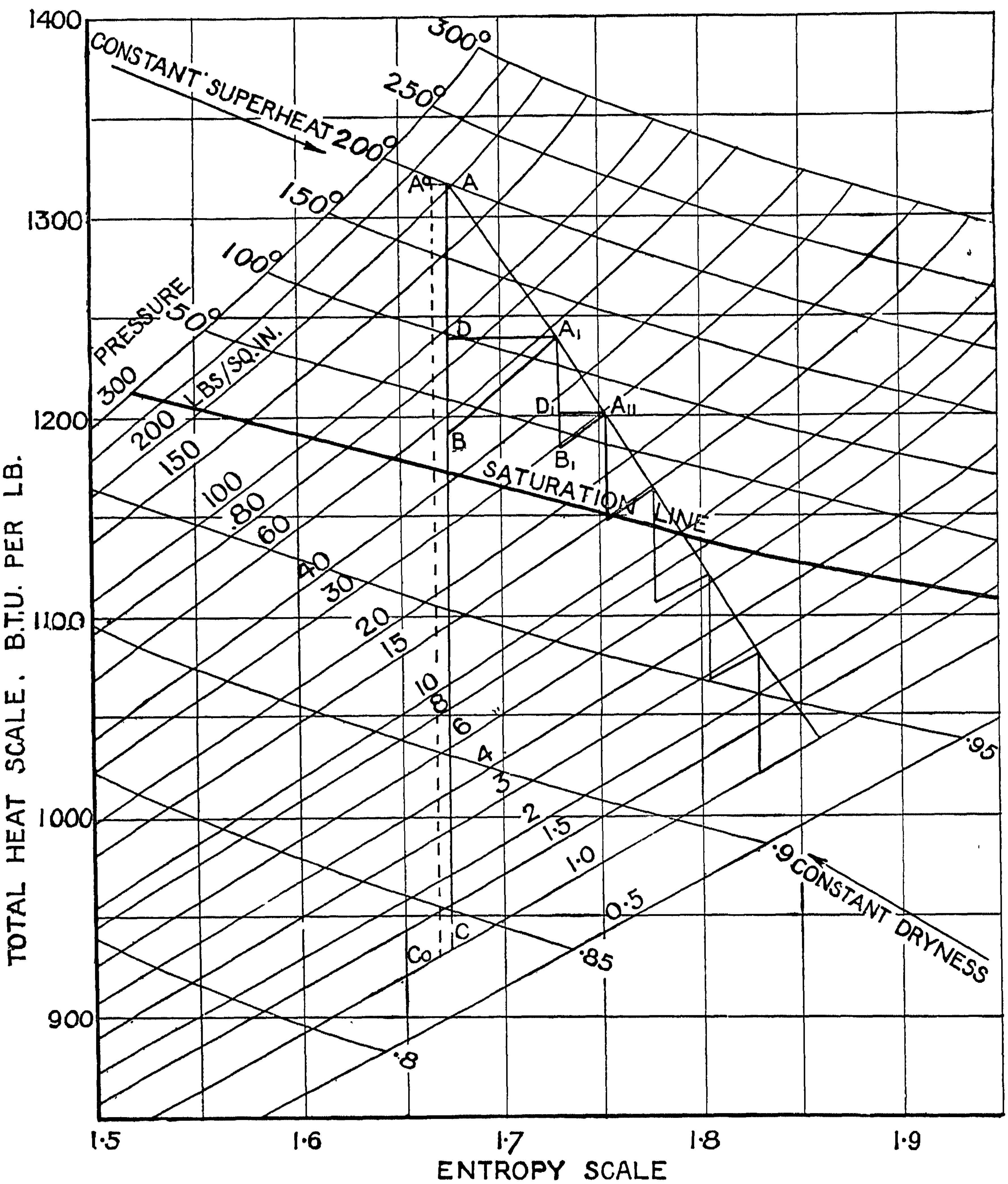


Fig. 2.—Mollier or Total Heat Chart

Governor Valve Loss.—In all turbines some form of governing device is necessary, and to obtain the required control of the flow of steam a drop of pressure is essential across the governing valve or valves. This

pressure drop produces a loss of energy, since the steam expands through the valve without doing any useful work. Referring again to fig. 2, the isothermal expansion or throttling through the governing valve is represented by the line A_0A. So that in this case the steam is supplied to the throttle valve at a pressure of 215 lb. absolute (i.e. 200 lb. on the pressure gauge), but before reaching the nozzles of the first stage its pressure has been throttled to 200 lb. absolute.

As a basis of comparison of all turbines it is usual to take the heat drop based on the pressure in front of the throttle valve, and consequently the heat drop would be given by the line A_0C_0.

CHAPTER III

Calculation of Blade and Nozzle Dimensions

Expansion of Steam in a Nozzle.—When steam is allowed freely to expand in a nozzle, the available energy or expansion energy released performs work upon the mass of the steam itself and produces kinetic energy in the form of a high steam velocity.

Let P_1 be the initial pressure at inlet to nozzle in pounds per square inch;

P_2, the final pressure at discharge from nozzle in pounds per square inch;

h, the heat available in adiabatic expansion between the pressures P_1 and P_2 in B.Th.U.;

W, the weight of steam passed per second;

C_0, the theoretical velocity attained at discharge from the nozzle, assuming no frictional loss (in feet per second).

Then the energy available is WJh (ft.-lb.),

And the kinetic energy generated is $W\frac{C_0^2}{2g}$ (ft.-lb).

Equating these values, we have

$$WJh = W\frac{C_0^2}{2g};$$

whence

$$C_0^2 = 2gJh,$$

$$C_0 = 223{\cdot}8\sqrt{h}, \qquad (1)$$

This expression gives the full theoretical velocity due to the heat drop h, assuming that there is no loss due to frictional resistance, and that in front of the nozzle the steam is initially at rest.

In actual practice the velocity obtained is always less than the theoretical. Thus, if C_1 is the actual velocity,

$$C_1 = 223{\cdot}8\phi\sqrt{h}. \qquad (2)$$

where ϕ is the velocity coefficient of nozzle.

With a correctly constructed nozzle the value of ϕ is comparatively high, experiments having shown values varying from about 0·94 to 0·97.

In actual practice the mean value of 0·955 probably represents a fairly satisfactory figure.

As will be fully explained later, where the heat drop is great, to approach the full theoretical velocity a divergent or expanding nozzle shape is required.

Critical Pressure Ratio.—Consider a simple nozzle as shown in fig. 3 connecting two closed chambers in which exist steam pressures of P_1 and P_2. At first let these two pressures be equal; equilibrium will then exist, and no flow will take place through the nozzle. If now, whilst maintaining the initial pressure P_1 at a constant value, the pressure P_2 be decreased, steam will begin to flow through the nozzle, and as the discharge pressure P_2 is gradually reduced the quantity of steam passed by the nozzle will increase.

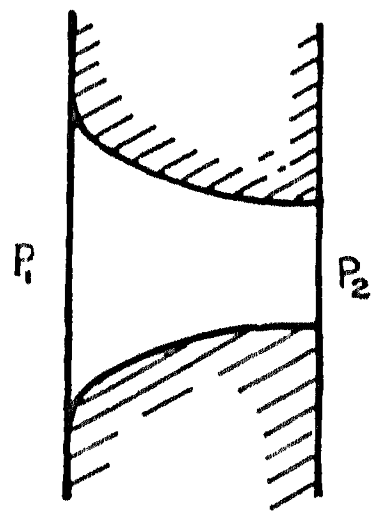


Fig. 3.—Simple Nozzle

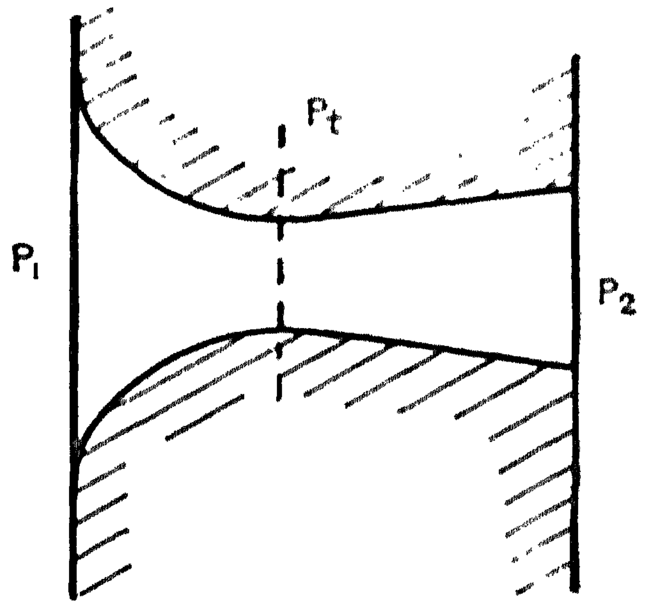


Fig. 3A.—Divergent Nozzle

At first as the discharge pressure falls the quantity of steam passed will increase rapidly, but as the pressure continues to fall the rate of increase of the discharge diminishes, and when the ratio of the pressures $P_2 : P_1$ reaches a value between 0·5 and 0·6 the discharge quantity will reach a maximum value; any further reduction of P_2 will produce no further increase in the amount of steam flowing. This particular pressure ratio at which the maximum discharge is reached is known as the critical pressure ratio, and when this ratio exists the velocity of the steam in the throat of the nozzle is equal to the speed at which sound would be transmitted through the steam.

If to the simple nozzle (fig. 3) a suitable divergent discharge portion be added as in fig. 3A, the conditions now existing will be that at the throat of the nozzle a pressure P_t corresponding to the critical pressure will exist, and beyond this in the divergent portion of the nozzle there will be a further expansion of the steam down to the final discharge pressure P_2. Thus although at the throat of the jet it is impossible to obtain a greater velocity than the speed of sound, by employing a suitably divergent nozzle a final discharge velocity corresponding to the full pressure drop may be obtained.

The simplest mathematical proof of the theory of critical pressure is afforded by considering the operation of the Rankine cycle referred to pressure-volume diagram.

In fig. 4 the vertical and horizontal axes represent respectively the pressure and volume scales of 1 lb. of steam. BC represents the increase in volume when 1 lb. of water is converted into steam at a pressure P_1. CD represents the adiabatic expansion line from the initial pressure P_1 and specific volume V_1 to the final pressure P_2 and specific volume V_2. DA represents the reduction in volume as the steam, at pressure P_2, is condensed back to water. The total work available during the cycle is represented by the area ABCD.

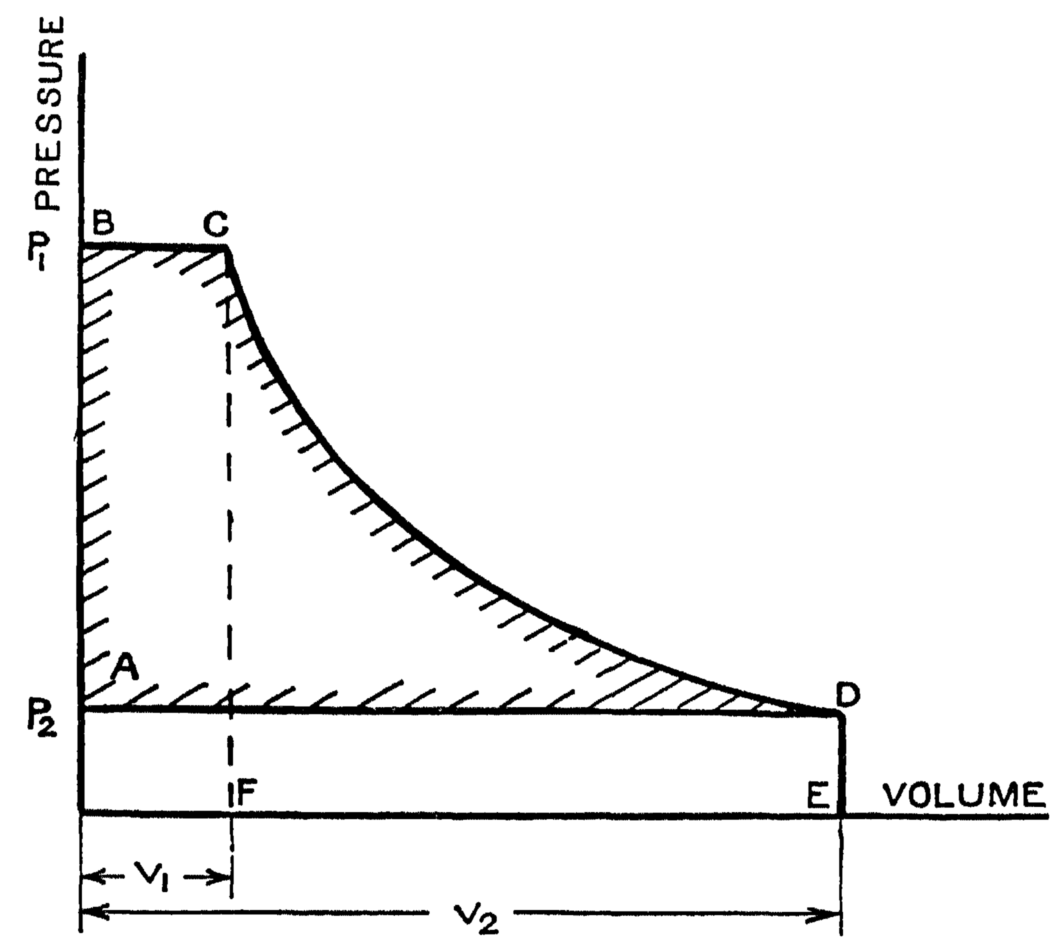


Fig. 4.—Rankine Cycle PV Diagram

During adiabatic expansion the pressures and specific volumes of steam are very accurately represented by a law of the form

$$PV^n = \text{constant}.$$

The curve from C to D follows this law, and area CDEF under this curve is given by the expression

$$\int_{V_1}^{V_2} PdV = P_1V_1\int_{V_1}^{V_2}\frac{dV}{V^n}$$
$$= \frac{P_1V_1 - P_2V_2}{n - 1}.$$

The expression for the work (W) available during adiabatic expansion can now be ascertained. It is given by

$$W = P_1V_1 + \frac{P_1V_1 - P_2V_2}{n - 1} - P_2V_2$$
$$= \frac{n}{n-1}\left[P_1V_1 - P_2V_2\right]$$
$$= \frac{n}{n-1}\left[1 - \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}\right]P_1V_1.$$

To obtain the value of the work available in foot-pounds, the pressures must be stated in pounds per square foot. In the nozzle this available work is converted into kinetic energy, $\frac{C_0^2}{2g}$, and thus

$$C_0 = \sqrt{\frac{2gn}{n-1}\left[1 - \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}\right]P_1V_1}.$$

If now A is the nozzle area in square feet, and W the quantity of steam passed in pounds per second,

$$C_0 = \frac{WV_2}{A}$$

$$= \frac{W}{A}V_1\left(\frac{P_1}{P_2}\right)^{\frac{1}{n}}, \qquad \text{since } P_1V_1{}^n = P_2V_2{}^n.$$

Thus
$$\frac{W}{A} = \frac{1}{V_1}\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}\sqrt{\frac{2gn}{n-1}\left[1-\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}\right]P_1V_1}$$

$$= \sqrt{\frac{P_1}{V_1}\frac{2gn}{n-1}\left[\left(\frac{P_2}{P_1}\right)^{\frac{2}{n}}-\left(\frac{P_2}{P_1}\right)^{\frac{n+1}{n}}\right]}.$$

Let $x = \frac{P_2}{P_1}$, then this expression, giving the weight passed per unit of area, reaches a maximum for some definite value of x.

Writing

$$y = x^{\frac{2}{n}} - x^{\frac{n+1}{n}},$$

$$\frac{dy}{dx} = \frac{2}{n}x^{\frac{2-n}{n}} - \frac{n+1}{n}x^{\frac{1}{n}}$$

$$= 0, \text{ when } y \text{ is a maximum;}$$

whence

$$x_c = \left(\frac{2}{n+1}\right)^{\frac{n}{n-1}} \text{ at the maximum point}$$

$$= \text{critical pressure ratio.}$$

For superheated steam the value of n is 1·3, and for saturated steam 1·135. These values give a critical pressure ratio of 0·546 for superheated steam, and 0·577 for saturated steam.

Inserting the value of the critical drop in the formula for discharge, a simple expression for discharge per square inch of nozzle area is obtained:

$$\frac{W}{a} = k\sqrt{\frac{P_1}{V_1}}, \ldots\ldots\ldots\ldots\ldots\ldots\ldots(3)$$

where a is the nozzle area in square inches;

P_1, the initial pressure in pounds per square inch;

V_1, the initial specific volume of steam in cubic feet per pound;

k, a constant; 0·315 for superheated steam, and 0·3 for saturated steam.

Many experiments have been carried out to ascertain the actual value of k as compared with the theoretical values stated above. In general, experimental evidence points to the fact that a value of $k = 0·31$ may be attained in a well proportioned nozzle, both for superheated and saturated steam. This means that the discharge of saturated steam exceeds the theoretical

value. The explanation of this appears to be that when saturated steam expands very rapidly it momentarily assumes an unstable or supersaturated condition in which the temperature falls below the saturation temperature corresponding to the pressure, but condensation does not fully occur.

In other words, the steam instantaneously behaves like superheated steam, and more or less follows the law $PV^{1\cdot3} =$ constant.

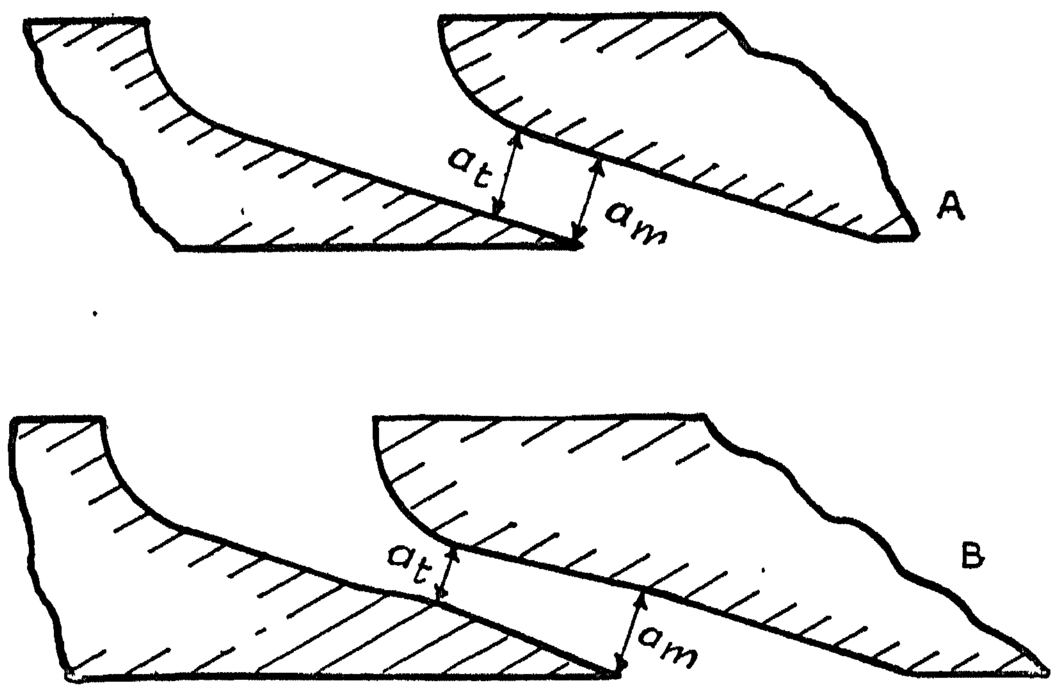


Fig. 5.—Nozzle Forms

Fig. 5 indicates the two forms of nozzles employed in practice.

A is nozzle employed where the ratio $\frac{P_2}{P_1}$ is not less than about 0·5. In this case (throat area a_t) = (mouth area a_m), and these areas are calculated from the velocity derived from equation (2), page 145.

B is nozzle employed where the ratio $\frac{P_2}{P_1}$ is less than 0·5. In this case (throat area a_t) is less than (mouth area a_m), and throat area is calculated by equation (3), p. 148, whilst mouth area by equation (2).

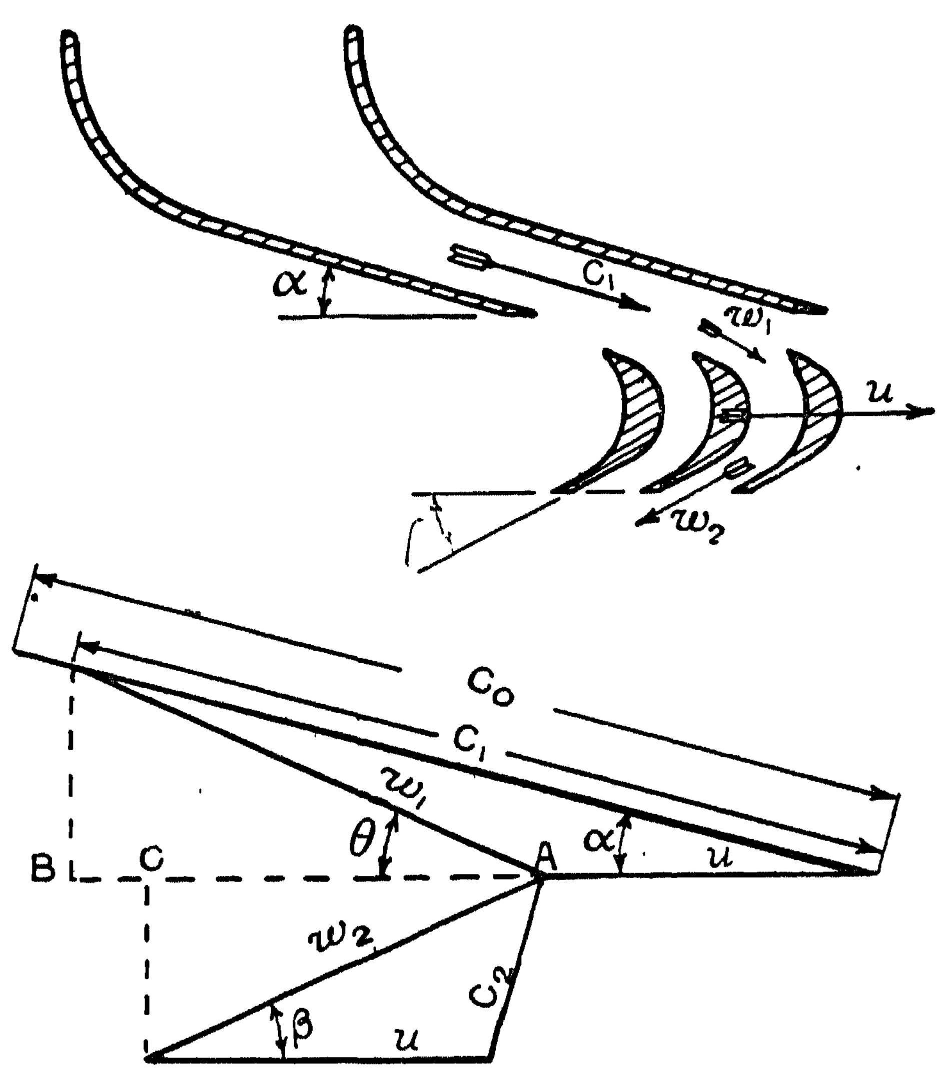


Fig. 6.—Velocity Diagram for Simple Impulse Turbine

The Impulse Blade.—In the impulse type of turbine the whole of the expansion energy of the steam is converted into kinetic energy in stationary nozzles from which the steam is directed into the moving blading, and consequently the passage through the moving blading occurs without pressure drop.

Fig. 6 shows a single stage of a simple impulse turbine, and the vector diagram representing the steam velocities.

Steam leaves the nozzle having an angle of inclination α at a velocity C_1. Subtracting from this the speed of the blading u, the relative velocity w_1 at which the steam enters the blading is obtained. In passing through the blading frictional losses reduce the velocity to w_2, and it leaves the blading with this relative velocity at an angle β equal to the discharge angle of the blading. Again, subtracting from w_2 the blade speed u, the absolute

velocity C_2 is obtained at which the steam passes away from the turbine stage.

From the vector diagram the driving force due to the steam can be calculated, and thence the efficiency of the stage ascertained.

The change of tangential velocity in the steam passing through the blading is given by the tangential components of w_1 and w_2; thus:

$$\text{Let the ratio of} \quad \frac{w_2}{w_1} = \psi$$
$$= \text{velocity coefficient of blading.}$$

Experiments show that the value of ψ for a well-shaped blade lies between the values of 0·8 and 0·9, according to the velocity of the steam and the angle of deflection in the blading.

Also let it be assumed that the angle β of blading is equal to angle θ at which the steam enters the blades, this being approximately true in most cases. Then

$$\begin{aligned}
\text{change of tangential velocity} &= \text{AB} + \text{AC} \\
&= (C_1 \cos\alpha - u)(1 + \psi); \\
\text{driving force per pound of steam} &= \frac{1}{g}(C_1 \cos\alpha - u)(1 + \psi); \\
\text{work done per pound of steam} &= (\text{driving force}) \times (\text{blade speed}) \\
&= \frac{u}{g}(C_1 \cos\alpha - u)(1 + \psi).
\end{aligned}$$

But work available in steam is that due to the theoretical steam speed C_0, $\frac{C_0^2}{2g}$ ft.-lb., and $C_1 = \phi C_0$.

$$\begin{aligned}
\text{Blading efficiency} &= \frac{2u(1 + \psi)(\phi C_0 \cos\alpha - u)}{C_0^2} \\
&= 2(1 + \psi)(\phi r \cos\alpha - r^2),
\end{aligned}$$

where r = ratio u/C_0.

For any definite values of ϕ, ψ, and α this curve takes the form of a parabola, and for usual values met in practice the efficiency reaches a maximum at a value of the ratio r of approximately 0·46. Fig. 7 shows at A a typical efficiency curve of simple impulse stage for varying values of r.

The Velocity Compounded Stage.—It will be seen from fig. 7 that where the ratio r is low the possible efficiency is also poor. To improve the efficiency where low values of r become necessary a velocity compounded wheel is employed. Referring to fig. 8, the steam on leaving the first row of moving blades at a velocity C_2 is deflected through a row of fixed return blades into a second row of moving blades. The velocity diagram for the complete stage is shown, and in this case the change of tangential velocity becomes the sum of the tangential components of W_1, W_2, W_3, and W_4, and the efficiency curve at the lower values of r is improved as indicated at curve

B (fig. 7). It will be seen, however, that maximum efficiency attainable with the velocity compounded wheel is not as high as for the simple impulse stage, and consequently where the highest possible blading efficiency is aimed at a turbine composed entirely of simple velocity stages is frequently adopted.

For many conditions, however, the two-row wheel has considerable practical advantages, since with a ratio of $r = 0{\cdot}23$ instead of 0·46 in the case of a single-row wheel, the amount of heat utilized per stage is four times

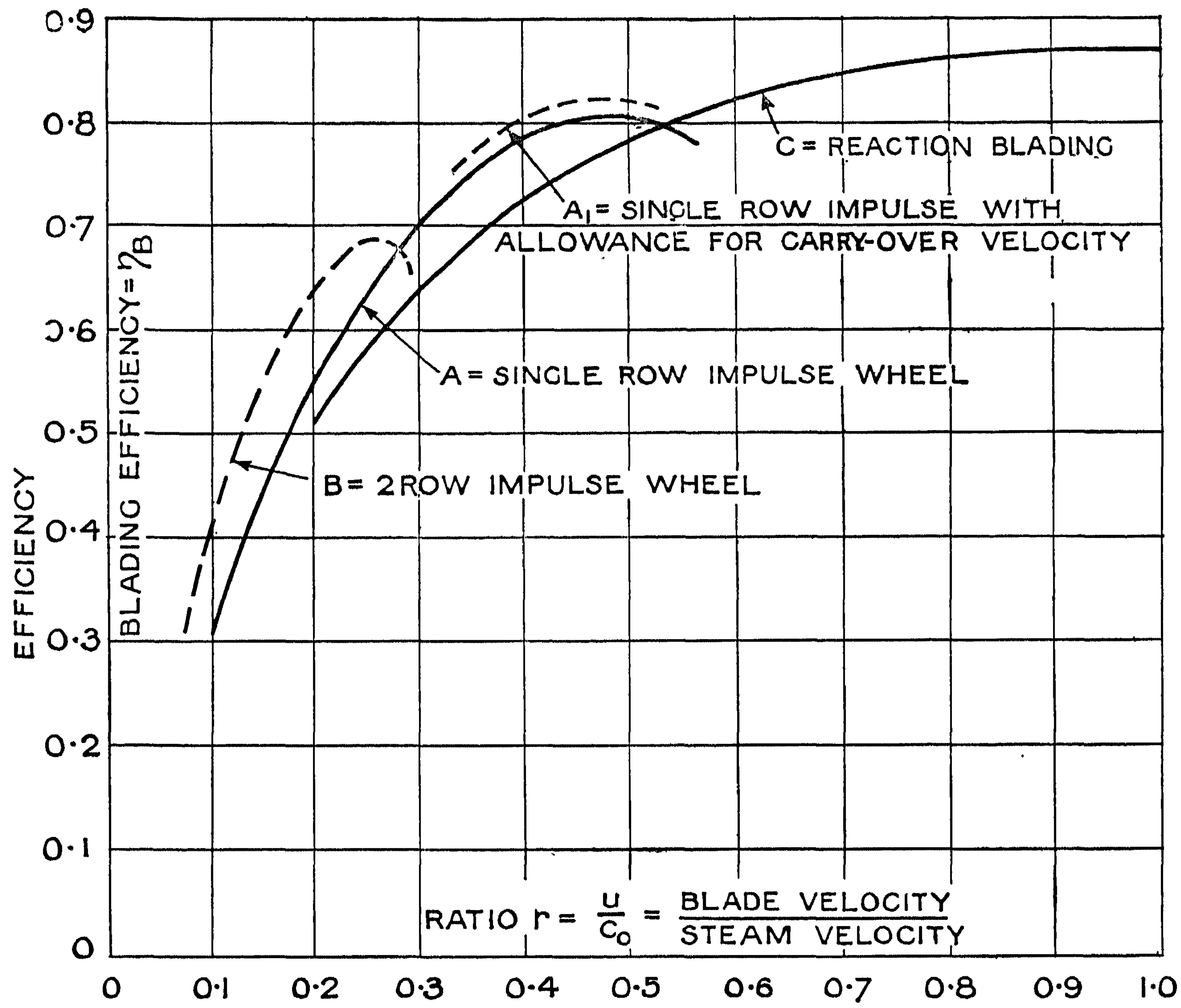


Fig. 7.—Efficiency Curves

as great, and consequently steam may be expanded down to a much lower pressure in the first stage; and in many cases the gain due to the lower leakage and steam frictional losses compensate for the lower blading efficiency.

Carry Over of Energy from Stage to Stage.—In the case of a multi-stage turbine a certain proportion of the residual velocity in one stage may be carried through to the succeeding stage and be there utilized. Referring back to fig. 6, the absolute velocity of discharge C_2 represents a certain amount of energy which may further be utilized in the turbine, and consequently as a true basis of comparison the blade efficiency may be credited with a certain amount of the carry-over energy. The effect on the efficiency curve of considering carry-over is shown at A_1, and, as will be seen later, this curve gives a fairer comparison with the reaction blading efficiency curve C.

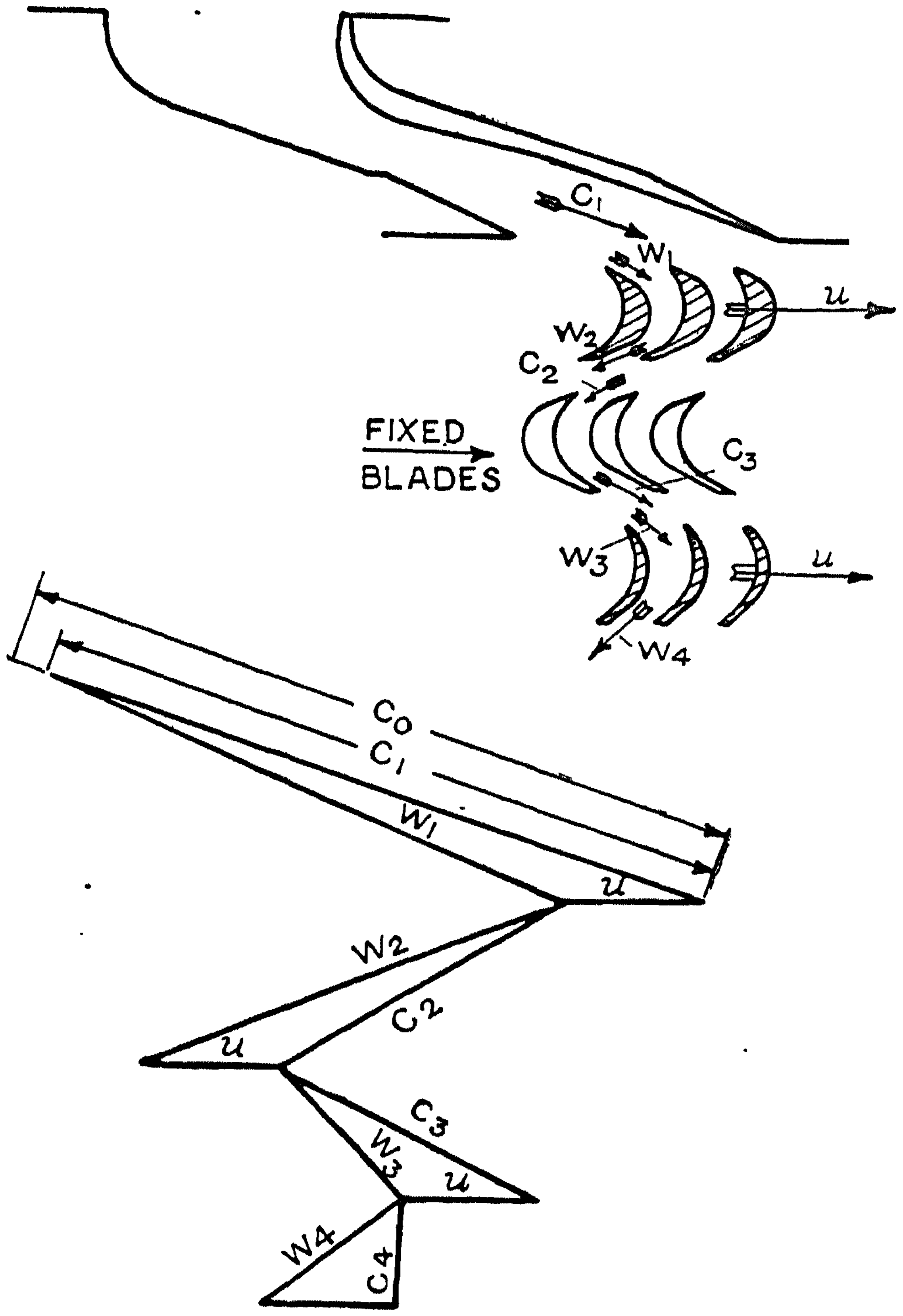


Fig. 8.—Diagram for Impulse Turbine Compounded for Velocity

Reaction Blading. – In the reaction type of turbine the expansion is divided almost evenly between both the fixed and moving blades. As has already been described (Chap. II), the blading of this type of machine is usually identical for both the fixed and moving elements. Also, as the expansion which occurs in each row is comparatively small the increase in volume of the steam is not great; it is thus usual practice to divide the blading into "expansions" or groups of several rows, the blading throughout each expansion being identical. It is thus obvious that as the steam expands through any group of blading its velocity must increase slightly from row to row. For most purposes it is usually sufficiently accurate to base the calculations on the mean specific volume of the steam in any group.

Fig. 9 shows the blading of a single stage comprising one fixed and one moving row. Steam enters the fixed blade at a velocity w. In passing through this blading a heat drop occurs and the velocity of discharge is C.

Thus if h = heat available in each row, assuming no frictional losses:

Available energy due to heat drop = increase of kinetic energy.

$$Jh = \frac{C^2}{2g} - \frac{w^2}{2g},$$

whence $C^2 = 2gJh + w^2$.

In an actual blade, however, as

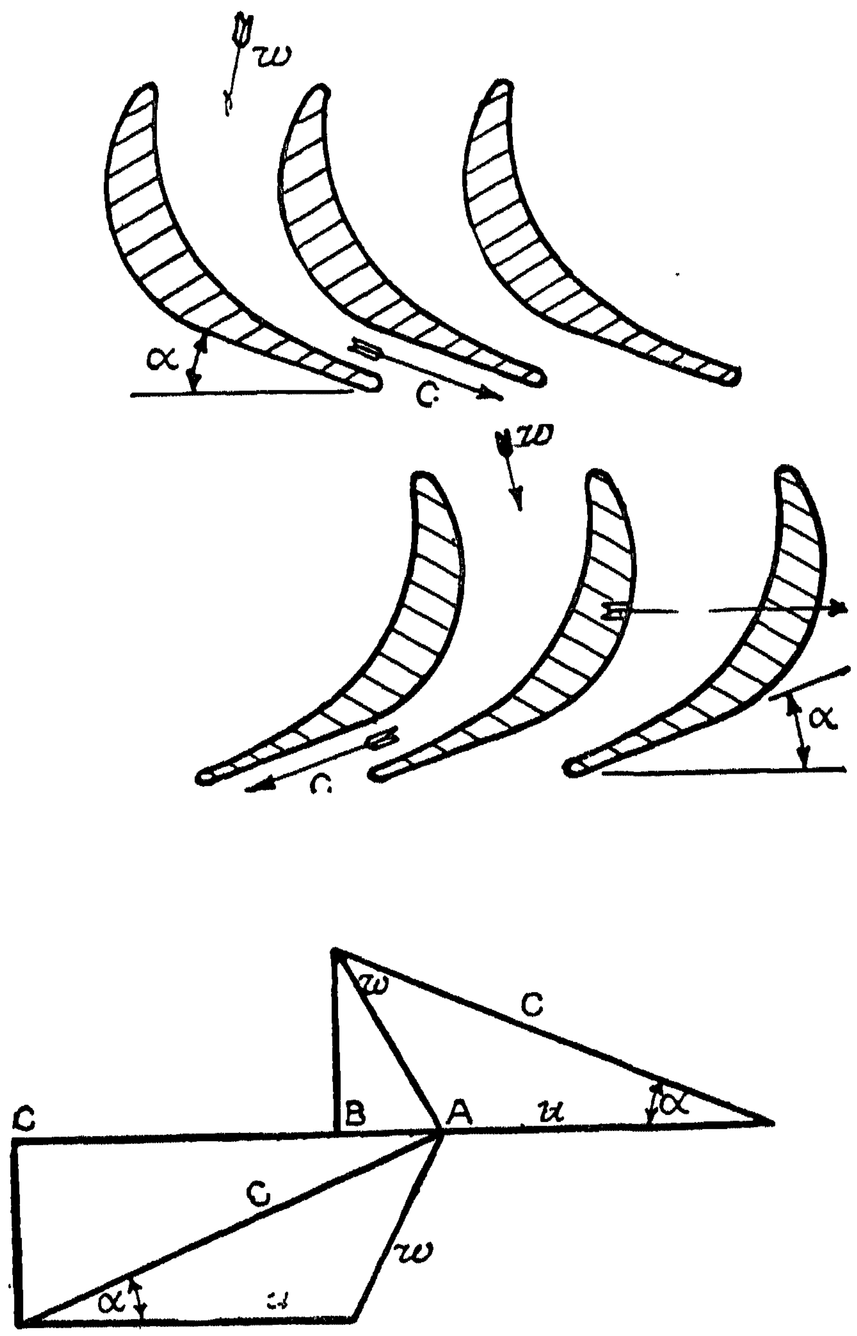


Fig. 9.—Velocity Diagram for One Stage of a Reaction Turbine

in a nozzle during the expansion of the steam, a certain amount of the energy is lost due to friction, and as in the impulse blade, a certain amount of the inlet velocity w is also lost. The actual equation thus becomes:

$$C^2 = 2gJh\phi^2 + w^2\psi^2 \dots\dots\dots\dots (4)$$

The coefficients ϕ and ψ are thus very closely allied to the coefficients for impulse nozzles and blading.

On leaving the fixed blading the relative velocity w at which the steam enters the moving row is obtained by subtracting the blade velocity u; and as it has been agreed to accept the mean value of the specific volume of steam, it is obvious that the velocities C and w are the same for both the fixed and moving blades. Consequently the velocity triangles are the same.

$$\left.\begin{array}{l}\text{Change of tangential velocity}\\ \quad\text{in moving blade}\end{array}\right\} = AB + AC$$

$$= 2C\cos\alpha - u;$$

$$\text{work done per pound of steam} = \frac{2Cu\cos\alpha - u^2}{g}.$$

Dividing this by the sum of the heat drops in the fixed and moving row, $2h$, we have the efficiency of blading:

$$\eta_B = \frac{2Cu\cos\alpha - u^2}{2gJh} \dots\dots\dots\dots (5)$$

To convert this expression into terms of the ratio $r = u/C$ from equation (4).

$$2gJh = \frac{C^2 - \psi^2 w^2}{\phi^2}.$$

Also for the velocity triangle

$$w^2 = C^2 + u^2 - 2Cu\cos\alpha.$$

Inserting these values in (5) gives an expression:

$$\eta_B = \frac{(2Cu\cos\alpha - u^2)\phi^2}{C^2 - \psi^2(C^2 + u^2 - 2Cu\cos\alpha)}$$

$$= \frac{(2r\cos\alpha - r^2)\phi^2}{(2r\cos\alpha - r^2)\psi^2 + (1 - \psi^2)}.$$

Taking mean values, we have $\phi = 0{\cdot}955$, $\psi = 0{\cdot}8$, and the normal blade angle of 20°, giving $\cos\alpha = 0{\cdot}9397$, and

$$\eta_B = \frac{1{\cdot}712r - 0{\cdot}91r^2}{1{\cdot}2r - 0{\cdot}64r^2 + 0{\cdot}36}.$$

The curve corresponding to this equation is shown at C in fig. 7. It will thus be seen that the maximum efficiency of reaction blading occurs at approximately twice the ratio for the maximum efficiency of the simple impulse stage shown at C; but since a pressure drop occurs in both fixed and

moving blades in a reaction turbine, it follows that for equal pressure drop in an element of one fixed plus one moving row the theoretical steam velocity is only $\frac{1}{\sqrt{2}}$ times that in a corresponding element of an impulse turbine. Therefore in a reaction turbine the blade speed to attain maximum efficiency is approximately $\sqrt{2}$ times that of an impulse turbine designed for its maximum efficiency for equal pressure drop across one element.

Further, it will be seen that curve C shows a higher maximum efficiency than curve A. Consequently it follows that a higher blading efficiency can be attained in the case of turbines embodying the reaction principle than in the case of those designed on the impulse principle. This advantage is not necessarily reflected in the over-all efficiency of the turbine, as losses other than those due to the blading constructions have an important bearing on the over-all efficiency.

In the case of reaction turbines, the pressure drop which occurs in both fixed and moving blades involves a leakage loss which generally exceeds the corresponding loss in an impulse turbine, which occurs in the clearance space between the diaphragms and the shaft.

Where dummy pistons are used for compensating the end thrust in reaction turbines, the steam leakage past these pistons involves another loss which does not occur in an impulse turbine. As a result it has been found in practice that high efficiencies have been obtained with turbines following both principles, and no decided advantage in the matter of economy has as yet been generally recognized for turbines involving one principle of blading over the other.

CHAPTER IV

Application of Steam Turbines on Land

During the past forty years the application of steam turbines has spread with remarkable rapidity, with the result that they have very largely superseded reciprocating steam-engines in many fields.

Their success has been mainly due to low capital cost; high thermodynamic efficiency, coupled with their capacity effectively to utilize a far greater steam range than can a reciprocating engine; and high space efficiency. In addition there are many minor advantages, such as low lubrication cost, absence of oil in their condensate, evenness of turning movement, and freedom from vibration. In the matter of reliability it is obvious that a breakdown of a machine such as a turbine, where the power transmitted for a given quantity of material is out of all proportion to that handled in a low-speed reciprocator, is apt to be of a comparatively more extensive nature.

It is true that there have been numerous serious failures of turbines, but these have mostly occurred in new and untried designs, and in view of

the extraordinarily rapid progress the number of such failures has been comparatively small. As far as well-tried designs are concerned, it is probably safe to say that steam turbines show at least equally favourable results in the matter of general reliability and maintenance costs as those obtained on other modern prime movers.

The non-reversibility of turbines, their comparatively poor economy when working non-condensing, and their essentially high speeds of revolution have retarded the progress of their application in some directions, such as the driving of locomotives, rolling-mills, &c.

The two vital desiderata, high efficiency and low first cost per unit of power, are practically functions of the speed of rotation and output. That is to say, for a given capacity the higher the speed the lower the cost and steam consumption for otherwise equal conditions, and again for any given speed the cost and consumption will be more favourable the greater the capacity.

The limits in this direction are not only set by the driven machinery, but also by the constructional difficulties in the building of the turbines themselves, which increase in proportion to both factors.

The curves shown in fig. 10 give an approximate indication of the present-day relation between maximum speeds and capacities for land turbines, and of corresponding figures for a period of about ten to twelve years ago, based on constructional considerations of the turbines alone. The difference between the two curves represents a measure of the progress of the turbine industry during that period. It will be noted that the later curve is prolonged in both directions. This is due to the growth in capacity for which the demand has arisen, and at the other end we have the result of the introduction of high-power gearing making the turbine designer to a large extent independent of the speed of the driven machine.

In deciding as to whether it is preferable to run a direct-coupled unit at a speed below the maximum permissible for the particular output to meet a fixed speed limit of the driven machine, or whether it is preferable to interpose gearing, the first cost and the power consumption of the gearing have to be set against the corresponding savings due to running the turbine at a higher speed.

The conditions under which it is advisable to interpose gearing between the turbine and the driven machine do not generally arise where the latter is a 50- or 60-cycle alternator or a blower or compressor. On the other hand, in the case of direct-current units, smaller 25-cycle alternators, pumps, &c., the reverse condition holds.

It should be noted that under certain conditions efficiency is of secondary consideration, e.g. where small turbines drive auxiliary plant, and where the exhaust heat is utilized for heating, &c. Under such conditions it is usual to build turbines of the one- or two-stage type, and to run them at such a speed as the driven machine may call for.

The steam turbine has probably found its most successful field of application as a prime mover of electric generators. Sir Charles Parsons at once realized the scope for his invention in this direction, and all his earliest

steam turbines were coupled to electrical generators. Up to that period electrical designers had been handicapped in the effective utilization of their materials by the comparatively low speeds demanded by reciprocating engines, and welcomed the advent of the steam turbine as a satisfactory means of overcoming this obstacle, although economical turbine speeds were far higher than those previously arrived at in electrical practice. In point of fact, the difficulties encountered in building direct-current turbo-generators proved

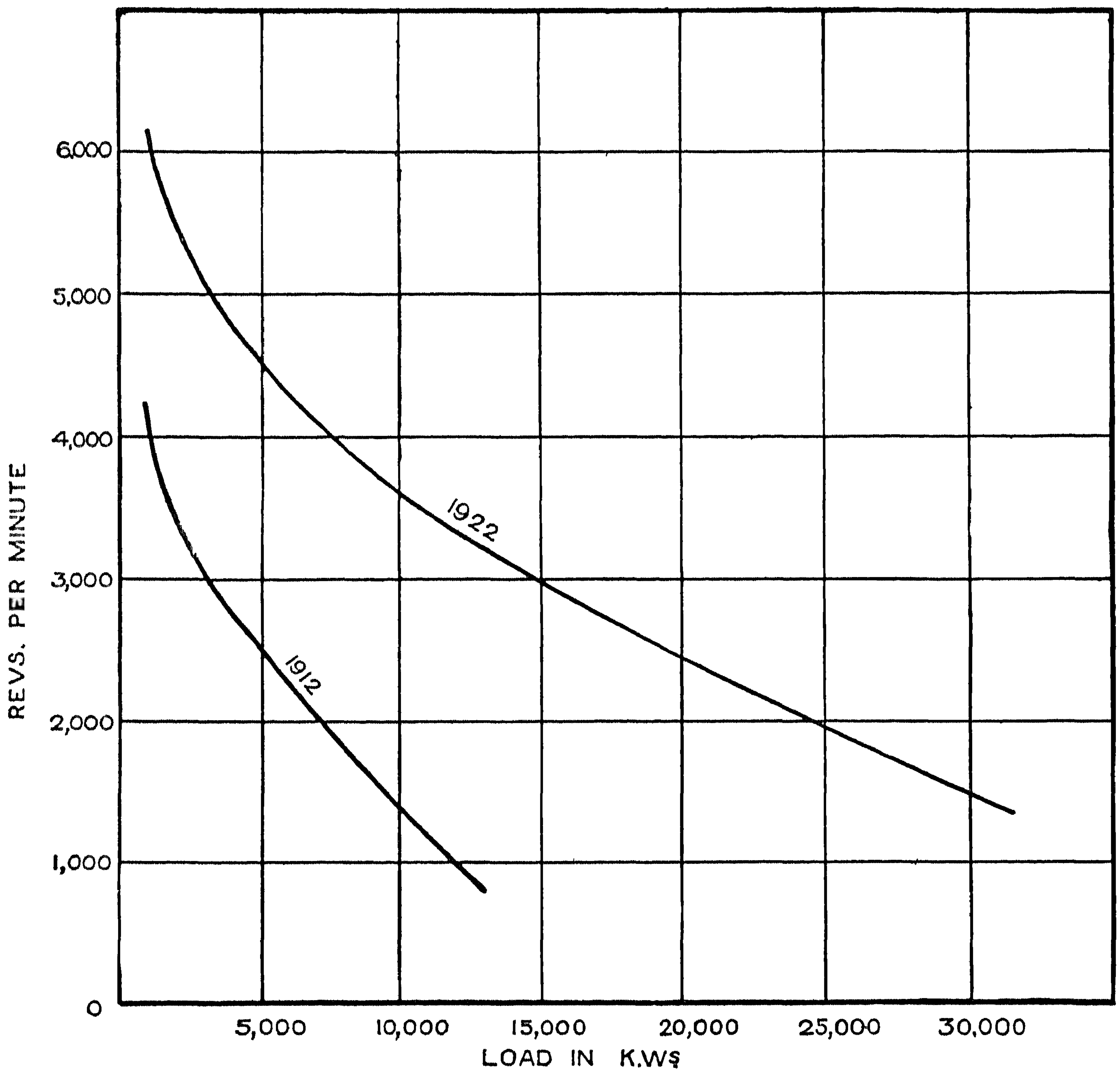


Fig. 10.—Maximum Speed Output Curves, 1912 and 1922

to be to all intents and purposes insurmountable in all but small sizes, and it was not until high-power turbine gearing was perfected that the success of direct-current turbine units was established.

On the other hand, the designers of alternators have succeeded in evolving a type of machine which differs radically in its mechanical characteristics from those adopted for low-speed designs, and which has enabled them to keep pace with turbine designers in their progress as defined by the speed output curve, except for the limitation imposed by the frequency.

There can be little doubt that the development of the steam-turbine-

driven alternator unit has been one of the most important factors in the growth of the electrical manufacturing and supply industries, and has made possible the evolution of modern steam-operated super-power stations.

Considerable progress has also been made in the development of turbine-driven blowers and compressors, most of the largest modern blower installations for steelworks and compressor plants for the coal- and gold-mining industries consisting entirely of turbine-driven units. On the other hand, the steam turbine has not yet succeeded in ousting the reciprocating engine in the large field of small-size blowers and compressors, owing mainly to the difficulty in constructing efficient small-power blowers and compressors of the high-speed rotary type.

CHAPTER V

The Parsons Type Turbine

The first practical steam turbine, the result of years of experiment and work, was built by Sir C. A. Parsons in 1884.

Fig. 11 is a section through a Parsons type compound reaction turbine, and shows features of the most up-to-date practice.

In the largest size turbines the necessity for the passage of large volumes of steam at the exhaust end has led to the splitting of the steam flow, allowing a double flow in opposite directions.

Two main diameters in the high-pressure portion and two in each of the low-pressure portions are made on the rotor. This follows the present-day practice of four main diameters in all.

Referring to fig. 11, steam enters at port 1 and flows through the turbine to the exhaust end 35. Immediately after the first expansion is the overload by-pass inlet belt 24.

The dummy pistons for balancing the rotor are shown at 19, 20, and 21, and the equalizing pipes for equalizing the dummy pressure and that on the corresponding portion of the rotor are shown dotted, 23 and 37.

The rotor body is forged in one piece. The top half casing fits over long upright standards in the lower half, so that it can readily and easily be removed and replaced. All high-pressure machines are fitted with an automatic atmospheric relief valve, so that should the vacuum fail at any time the turbine will exhaust to atmosphere.

The oil-tank will be seen together with the oil-pump, which supplies oil under pressure to all bearings.

Between the steam space and the bearings are placed the oil baffles and then the carbon segment glands—these are shown at 16, 30 and 17, 29.

The glands are steam packed, and 18, 28 show the vapour pipes from these glands.

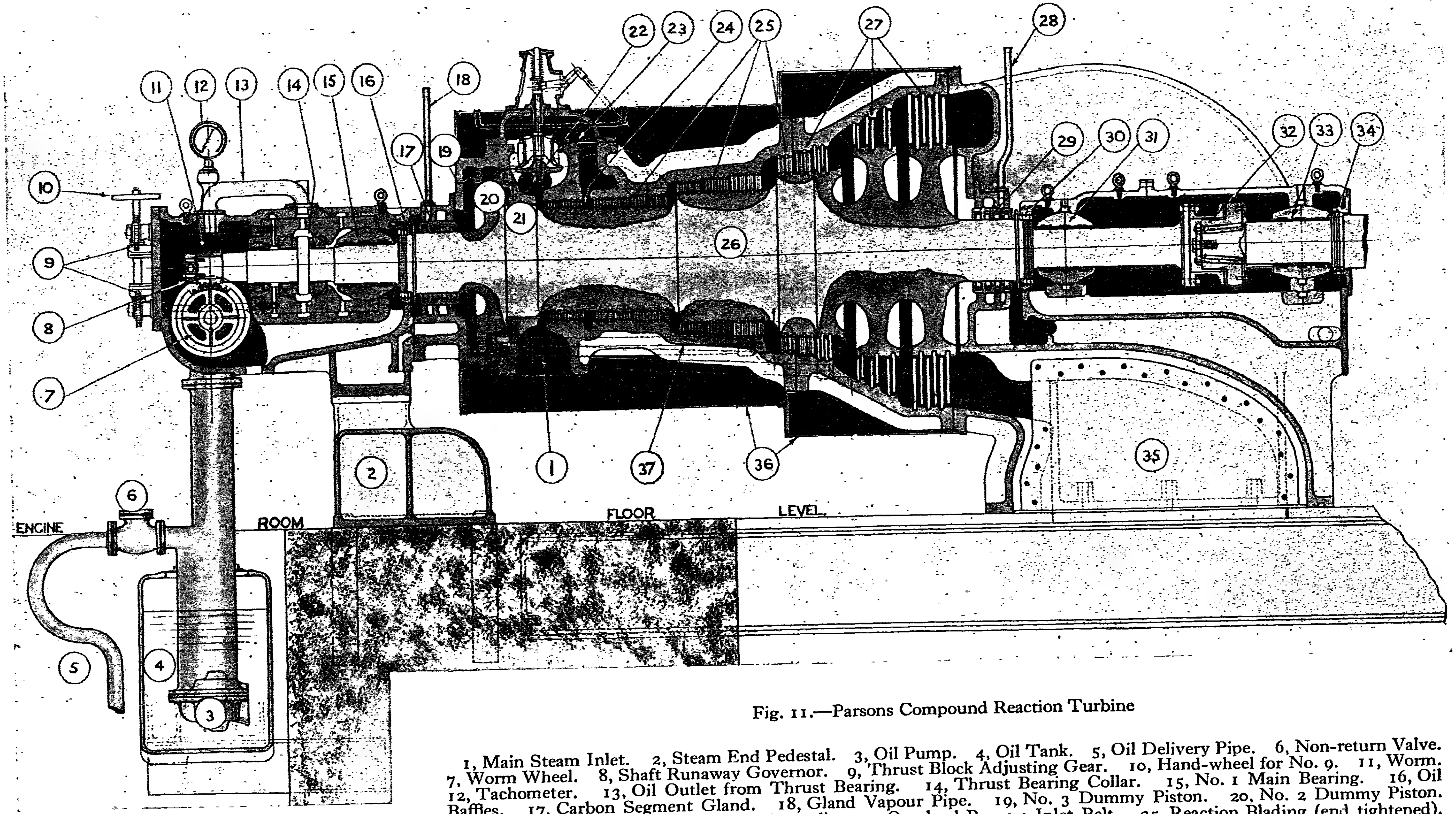


Fig. 11.—Parsons Compound Reaction Turbine

1, Main Steam Inlet. 2, Steam End Pedestal. 3, Oil Pump. 4, Oil Tank. 5, Oil Delivery Pipe. 6, Non-return Valve. 7, Worm Wheel. 8, Shaft Runaway Governor. 9, Thrust Block Adjusting Gear. 10, Hand-wheel for No. 9. 11, Worm. 12, Tachometer. 13, Oil Outlet from Thrust Bearing. 14, Thrust Bearing Collar. 15, No. 1 Main Bearing. 16, Oil Baffles. 17, Carbon Segment Gland. 18, Gland Vapour Pipe. 19, No. 3 Dummy Piston. 20, No. 2 Dummy Piston. 21, No. 1 Dummy Piston. 22, Overload By-pass Double-beat Valve. 23, Equalizing Pipe (dotted). 24, Overload By-pass Inlet Belt. 25, Reaction Blading (end tightened). 26, Turbine Shaft. 27, Reaction Blading (ordinary radial clearance). 28, Gland Vapour Pipe. 29, Carbon Segment Gland. 30, Oil Baffles. 31, No. 2 Main Bearing. 32, Flexible Coupling. 33, No. 3 Main Bearing. 34, Oil Baffles. 35, Turbine Exhaust. 36, Cylinder Lagging. 37, Equalizing Pipe (dotted). 38, Cylinder Cast-steel Centre.

The steam is admitted to the turbine through a governor valve. Both this valve and the overload by-pass valve are automatic, and are controlled from the governor through steam relays.

The relay for the governor valve has a small plunger valve as its exhaust valve in the relay cylinder. This plunger valve has an oscillating motion given to it from the spindle actuating the oil-pump, and is also directly controlled by the governor. Steam is constantly admitted to the relay cylinder. When the exhaust valve is shut, the steam pressure lifts the governor valve against the force of a spring. When the relay governor exhausts, this spring closes the governor valve. As the motion of the relay exhaust valve is oscillatory the main valve also alternately opens and closes, and hence the steam is admitted in gusts. These gusts become of longer and longer duration until at the overload capacity the admission is nearly continuous.

One of the most important improvements made in the detail of the Parsons machine is what is known as the end-tightened blading.

For many years Parsons turbines were made with small radial clearances between the tips of the blades and the surfaces of the cylinder and shaft. These fine clearances, which were necessary to prevent leakage of the steam over the blade tips, constituted the principal weakness of machines of the Parsons type.

The tip leakage was most serious at the high-pressure end of the machine, as the clearances there bore an appreciable relation to the length of the blades, which were here short. Moreover, it was just at this point where the higher temperature of the steam rendered it unwise to run the clearances too fine. Thus these radial clearances could not be reduced without endangering safety of operation, nor increased without lowering steam economy.

This disability has been minimized by the development of the present Parsons "end-tightened" blading. This has now been in commercial use about ten years, and is therefore established as regards reliability. This system of blading entirely eliminates fine radial clearances, appears to suffer very little deterioration with service, and gives mechanical reliability.

In fig. 12 a perspective view of end-tightened reaction pairs of rows (one fixed and one moving) forming a part of the blading of a Parsons machine is shown. It will be noted that whereas in the case of the original radial-clearance blading the spacing pieces at the roots of the blades are finished flush with the surface of the turbine cylinder and shaft, in the case of the end-tightened blading they project above and form a continuous barrier BB (see fig. 15).

The shrouding strip round the outer circumference of the blade rings projects over the edge of the blades on one side in such a manner that in each pair of rows the shrouding strip of one row projects against the barrier formed by the spacing strip of the other.

The space between the projecting shrouding strip and the corresponding barrier forms the working clearance which prevents the steam passing freely from one row of blades to the next.

It will be noted that this clearance is axial, and it can be adjusted at will

to any desired amount whilst the machine is running. That is, that when the machine has warmed up, the clearance can safely be made smaller. The radial clearances between the shrouding and the surface of the cylinder and shaft are made of the order of $\frac{3}{16}$ to $\frac{1}{4}$ in. or even more in large machines.

When a turbine is on load and has got heated up, the shaft expands longitudinally rather more than the cylinder, and since the thrust bearing collar at the high-pressure end may be looked upon as the point of tie between

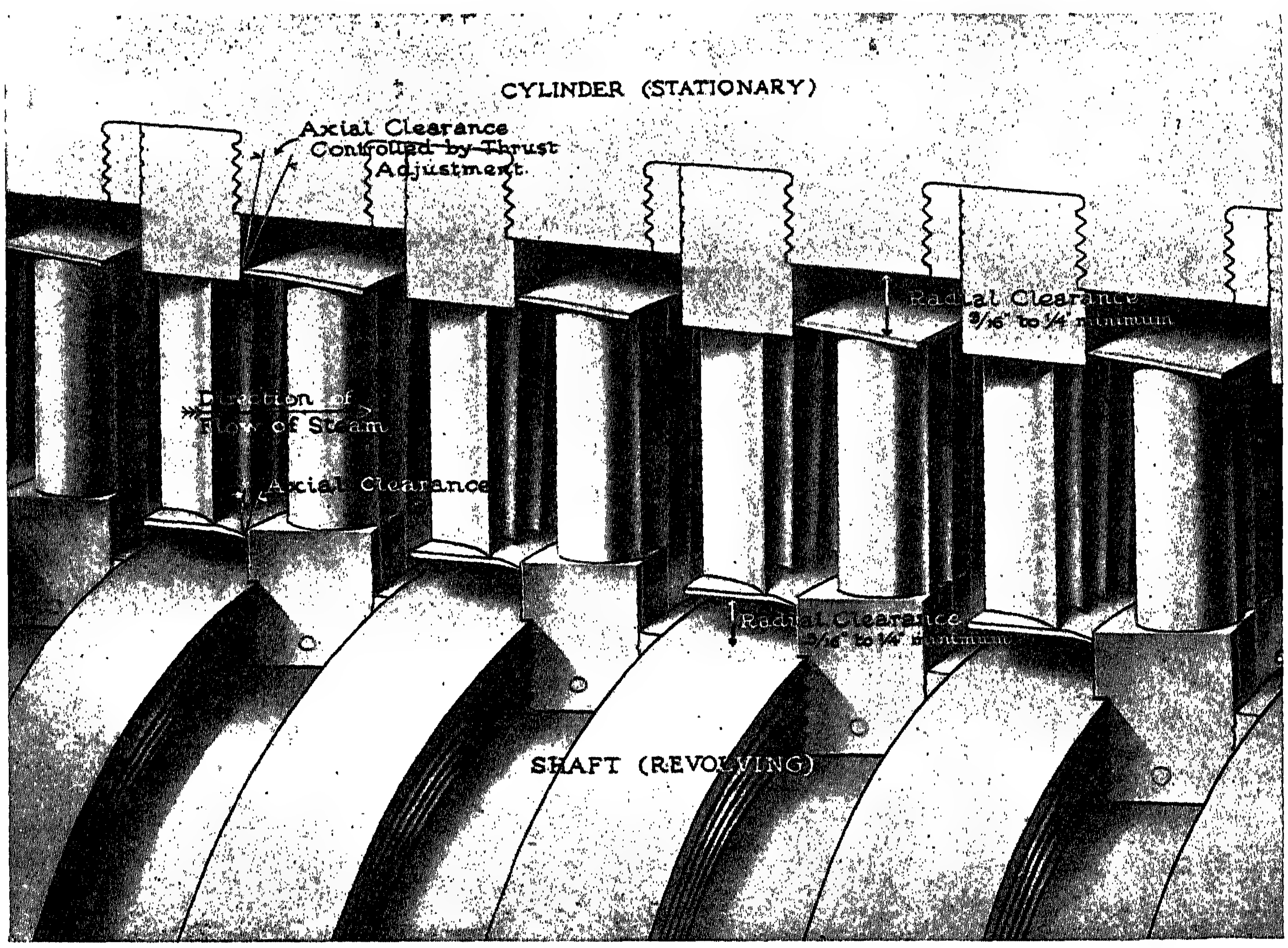


Fig. 12.—Parsons "End-tightened" Blading in Cylinder and Shaft, showing large Radial and Axial Clearance

the two, or the point of origin of the differential expansion, it will be realized that the tendency is for the axial working clearances to increase towards the exhaust end of the turbine, and hence the possibility of the binding together of the shrouds is to a certain extent obviated. Furthermore, the value of the end-tightened blading is greater at the high-pressure end where the steam leakage would be most serious. Thus larger radial clearances are obtained without increasing steam leakage, and greater steam economy can be realized with this type of blading than with the older forms.

The earliest designs of Parsons turbines had axial adjustment on working clearances in the dummy or balancing pistons and in the labyrinth glands. The same principle has been used with the end-tightened blading. All these axial working clearances are adjusted simultaneously. The position of the shaft referred to the cylinder is fixed by means of the thrust bearing.

This thrust bearing, of the adjustable pivoted type, is shown in longi-

tudinal section in fig. 13. The bearing carriage A, which is split horizontally but rigidly bolted together, can be moved slightly in a longitudinal direction carrying the turbine shaft with it. This movement is obtained as follows: steel rods B, fitted with collars C bearing on the bridge, are screwed into the carriage. The other ends of these rods are brought through the turbine casing and are connected together by a suitable gearing operated by a handwheel D and worm. This rotates both rods simultaneously by the same amount. By this rotation of the rods the bearing carriage can be moved through a short travel longitudinally in either direction as may be desired. The extent of the possible movement in either direction is limited by liners.

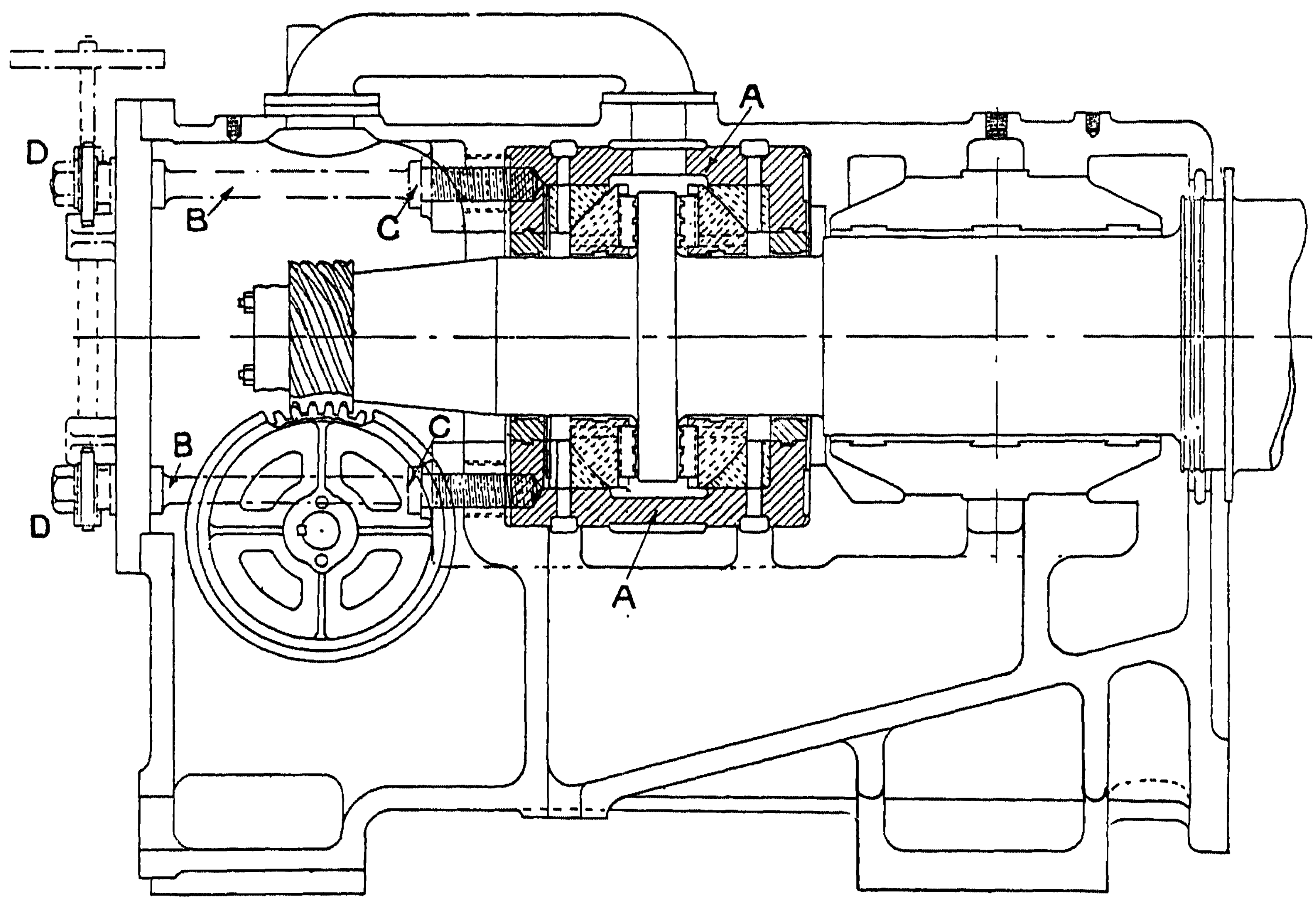
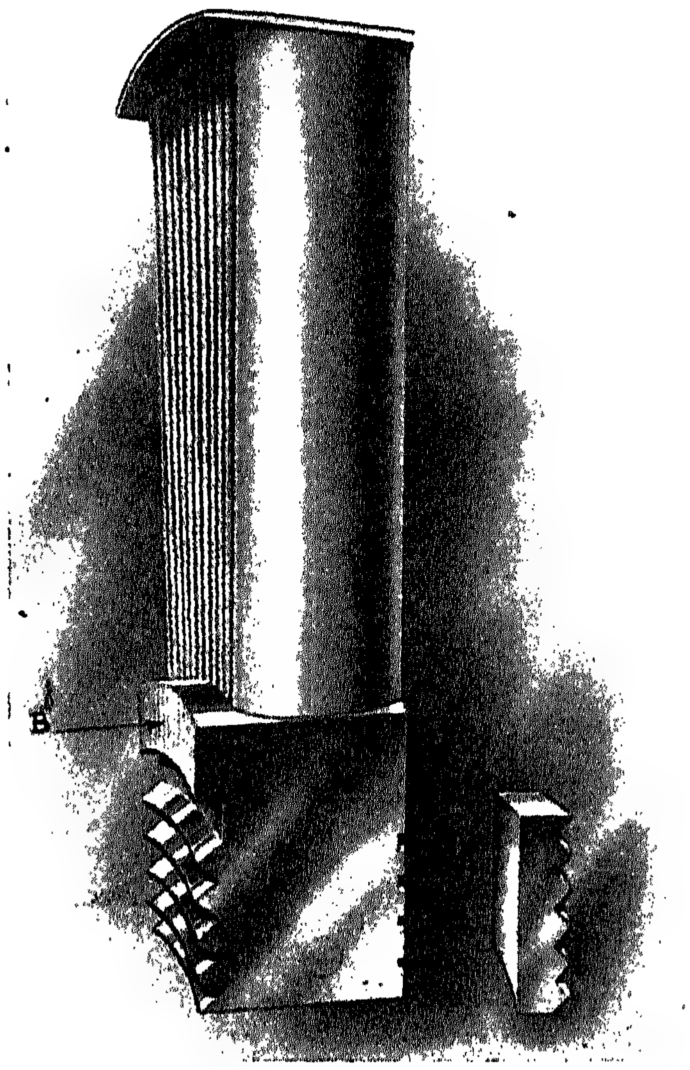


Fig. 13.—Parsons Steam Turbines. Longitudinal Section through Adjustable Pivoted Thrust-bearing

It cannot be assumed that this method entirely overcomes the blade clearance difficulty, for wherever non-lubricated running clearances are required there will be danger of touching, and consequent damage. At the same time, experience has proved conclusively that the end-tightened system has considerably reduced the risk and increased the reliability and economy of this type of turbine.

Method of Blade-fixing now Used.—Originally, and for many years, the Parsons blading was inserted alternately with distance pieces known as caulking pieces. These were made just to fit between the blade and blade-groove wall. The final operation was vertical caulking. Figs. 14 and 15 show views of built-up "blade units" of present-day Parsons reaction blading. These are now held in position in the shaft or cylinder by means of circumferentially driven serrated locking pieces. The locking pieces are driven up individually one against the other, each piece being swelled up so as to

completely fill all available space. The serrations machined on the blade units and locking pieces register with the serrated grooves in the shaft and cylinder, so that caulking is not now relied upon for securing the blades in place. Moreover, the method of fixing depends very little on the skill of the workman, as was the case with the older type of fixing.



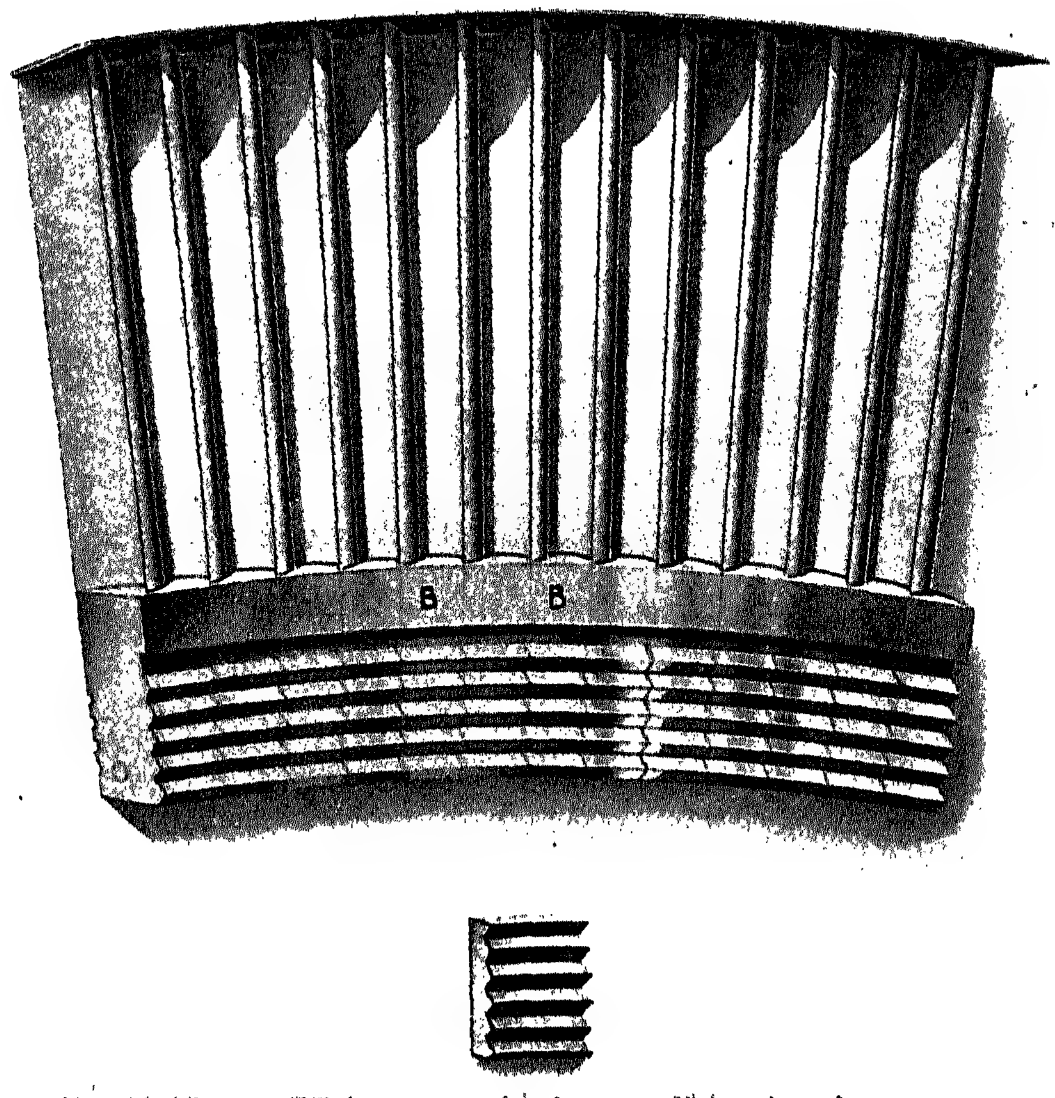


Fig. 14 Parsons "End-tightening Blading" Fig. 15

The shrouding strip used with end-tightened blading does away with the necessity for lacing strips except on the long blades in the exhaust end. The end-tightened device is not carried right through, and the low-pressure blading is the caulked-laced type.

These lacing strips are inserted at the inlet edge of the blades. On the longest blades two or even three may have to be used, but the tendency now is to make tandem machines for larger outputs, having a double flow at the exhaust end, and thus cut down the blade lengths and rotor diameters which would otherwise need special precautions for stiffening.

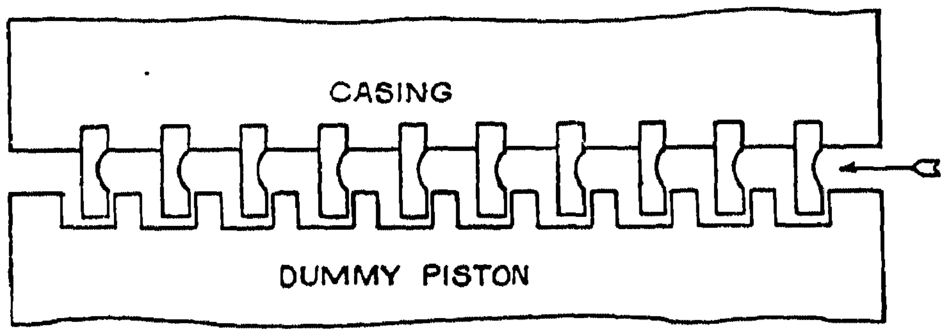


Fig. 16.—Radial Flow Labyrinth Packing

As will be seen by referring to fig. 11, the blades in the low-pressure end are mounted on discs, formed out of the body of the rotor forging. The blading is thinned at the tips, and the edges of the shrouding in the high-pressure portion, which may come in contact, can only touch along a thinned edge. Moreover, this end-tightened blading, if it touches at all, comes in contact with the spacing blocks which project above the groove for that purpose, and does not come in contact with the more

vulnerable part of the blade system, namely, the free portion of the blades.

The dummy pistons each have a diameter equal to the mean diameter of the blading on that section of the turbine they are to balance. To prevent steam leakage past them they have what is known as a "labyrinth packing". This is shown in detail in fig. 16.

The general practice in order to avoid strains on the turbine casing from expansion of the condenser body is to mount the condenser on springs. The condenser can then be bolted direct to the exhaust flange of the turbine, and expands downwards on to the springs when heated up.

CHAPTER VI

Rateau Turbines

FRASER & CHALMERS' DESIGN

Messrs. Fraser & Chalmers were the first builders of multi-stage impulse turbines in this country. Their original designs were built under licence from Professor Rateau, and were of the pure multi-pressure stage type. Later they, in common with many other turbine builders, adopted a velocity wheel for the first stage where conditions warranted its use, but they adhere to the pure pressure-stage design if steam conditions, &c., are such as to produce a more favourable design with this arrangement. Fig. 17 shows a 20,000-Kw. turbine running at 1500 r.p.m. having one velocity wheel, and single pressure stages for all lower stages. The last stages are arranged on the double-flow principle to avoid the excessively large disc diameters and the blade lengths which would otherwise be required efficiently to handle the steam at the high vacuum which this machine is intended to utilize. It will be seen that after the eleventh stage the steam path is divided, part of the steam passing through three more stages to the condenser, the remainder being carried by means of volutes and connecting passages to a two-stage system, the discs of which are of considerably greater diameter than those of the three stages with which they work in parallel. The profile of these two discs is so designed as to keep the factor of safety as high as in the three smaller discs. The diameter is, of course, still far below that which would be required in a corresponding single-flow design.

The high-pressure end of the casing is made of cast steel where the temperature of the steam warrants its use. Otherwise cast iron of special high-grade and homogeneous quality is employed. The exhaust end is fixed to the bedplate, and the high-pressure end is kept free to slide on guides to allow for heat expansion.

The discs are machined from solid forgings, and after the blades are assembled and the individual complete wheels are balanced, they are tested under steam in a special machine at a speed well above the normal. This

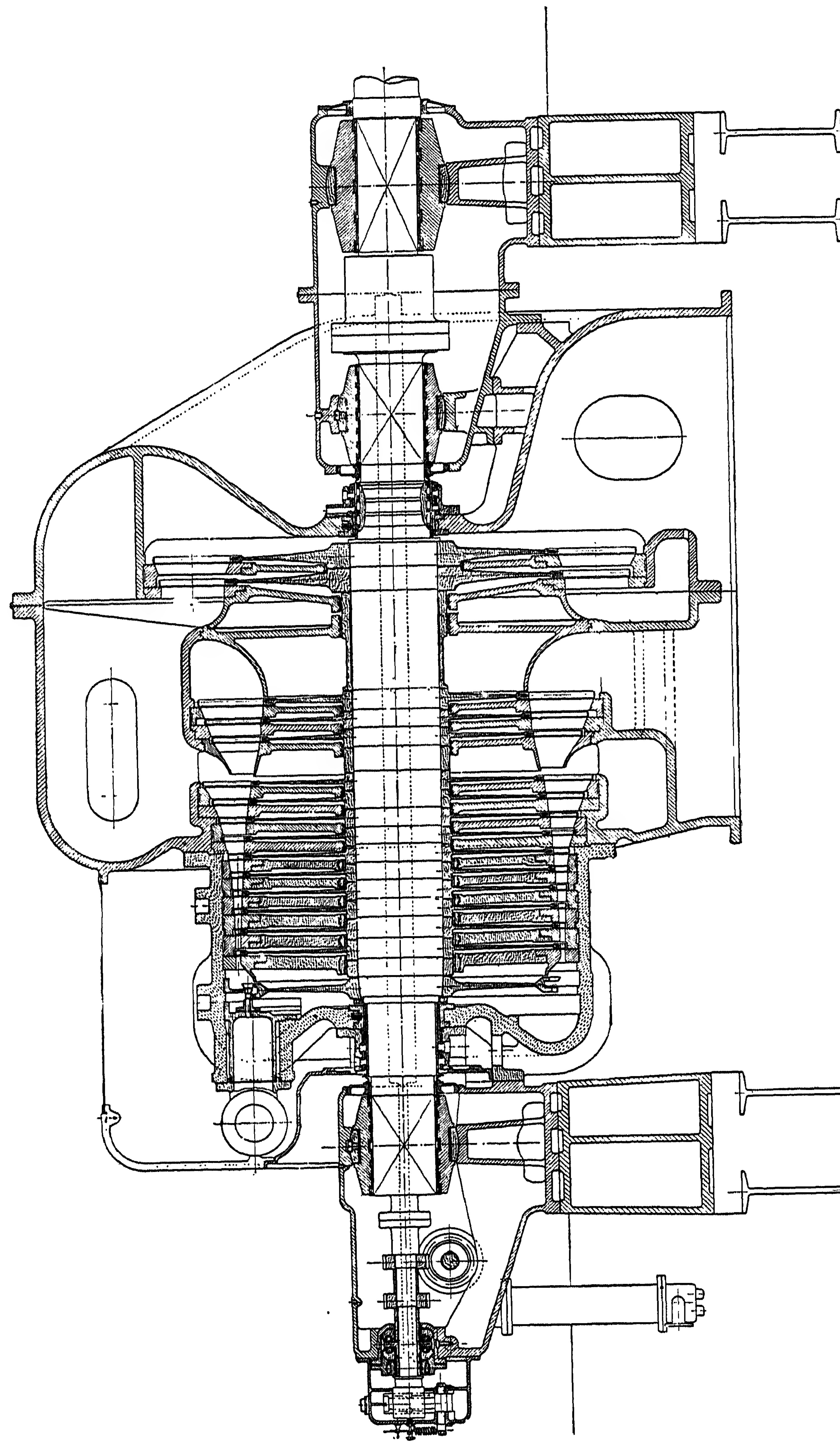
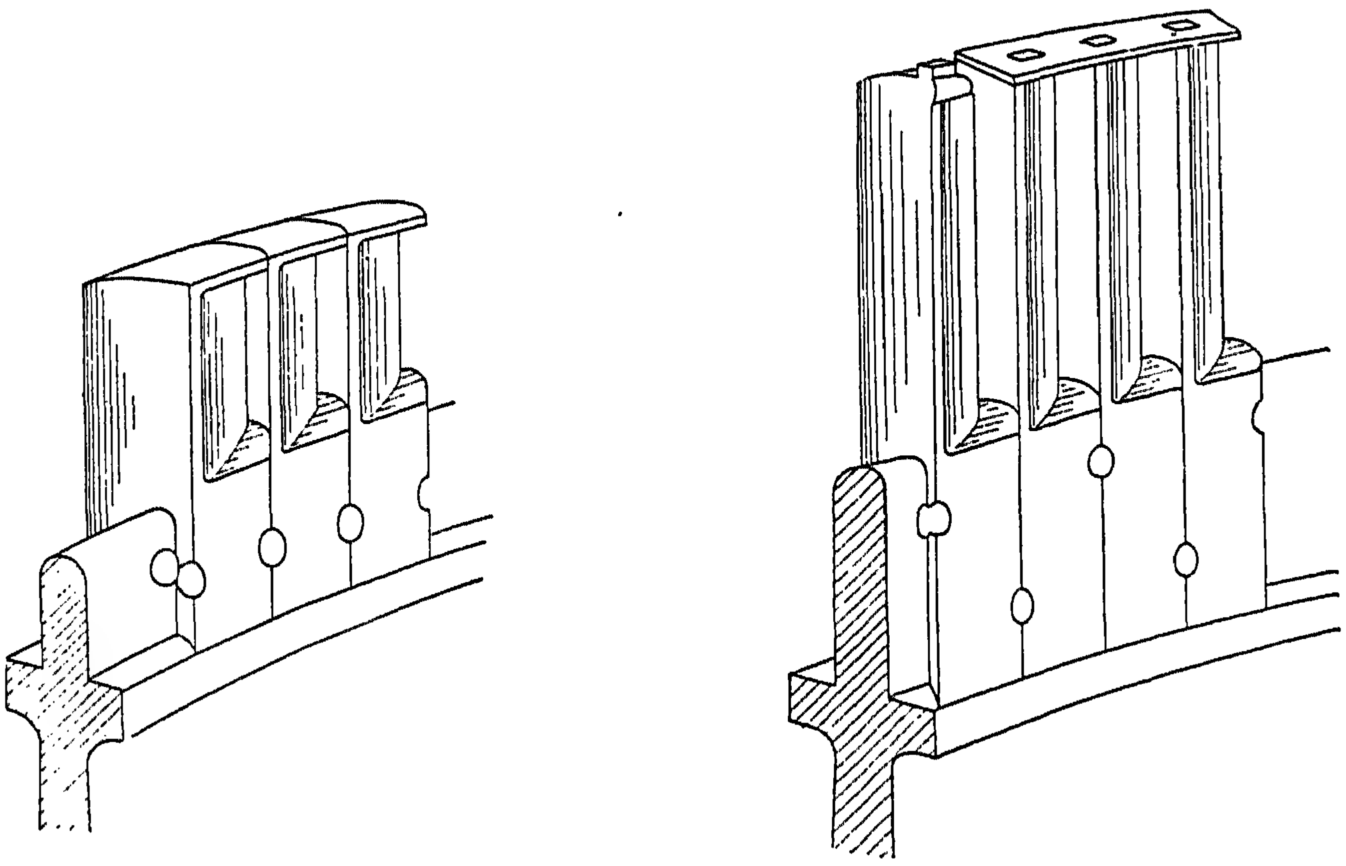


Fig. 17.—Sectional Arrangement of Fraser & Chalmers High-pressure Turbine. 20,000 Kw. Speed 1500 r.p.m.

practice is commended as a means of detecting defects such as hair cracks or unequal internal stresses or faulty workmanship in the fitting of the blades. The shafts are stepped from either end towards a maximum diameter approximately in the centre, and the discs are forced on and keyed in position. Suitable nuts are fitted at either end to lock the discs.

Fig. 18 illustrates Messrs. Fraser & Chalmers blade design and the methods of fastening. The blades are in all cases milled out of square bars, together with the forks which hold them to the disc. Each blade is so shaped at the root as to form its own distance piece, and in the shorter blades the outside



Blading for single or double rows (short blades). For re-blading, any blade can be taken off by removing the rivets on each side, and new blade can be riveted in place.

Blading for single row long blades. For re-blading, section of cover strip is taken off and blade then removed by taking out the rivets on each side.

Fig. 18.—Fastening of Blades of Fraser & Chalmers Rateau Turbine

distance piece or shrouding is also formed in one with the blades. In the case of the longer blades the ends are formed into tangs over which a band shrouding is fitted, the ends of the tangs being riveted over to hold the shrouding in position.

The blade forks form a straddle across the outer periphery of the discs, and the method of securing them to the discs by means of rivets passing through the forks and the discs is clearly shown in the illustration. This method of fixing the blades has the advantage of allowing individual blades to be removed and replaced in a simple manner.

Reverting to fig. 17 it will be noted that next to the journal is the worm driving the worm wheel attached to the main governor spindle, and on the other side of the worm the thrust block, which is of the Michell type. The emergency governor is fastened at this end of the turbine shaft, and consists of a rotating plunger whose centre of gravity is out of centre with the turbine shaft. The centrifugal force due to the out of balance of the bolt is checked by a spring until the predetermined tripping speed is reached. Beyond that point the plunger flies out and releases a trip lever and spring which

closes the main stop valve; at the same time the oil supply to the relay system is interrupted and opens a drain on the oil-relay cylinders, admitting also the oil supply to the top of the oil-power piston. The throttle valve is therefore closed simultaneously. Provision is made to enable the emergency gear to be tripped by hand. The oil-pump is placed near the floor, and is attached to the bearing housing by a special column which makes the pump quite accessible. The pump is of the rotary type, driven from the main governor spindle by gears. The oil coming from the bearings runs into a tank in the bedplate and passes through a strainer into the pump suction. The oil is then

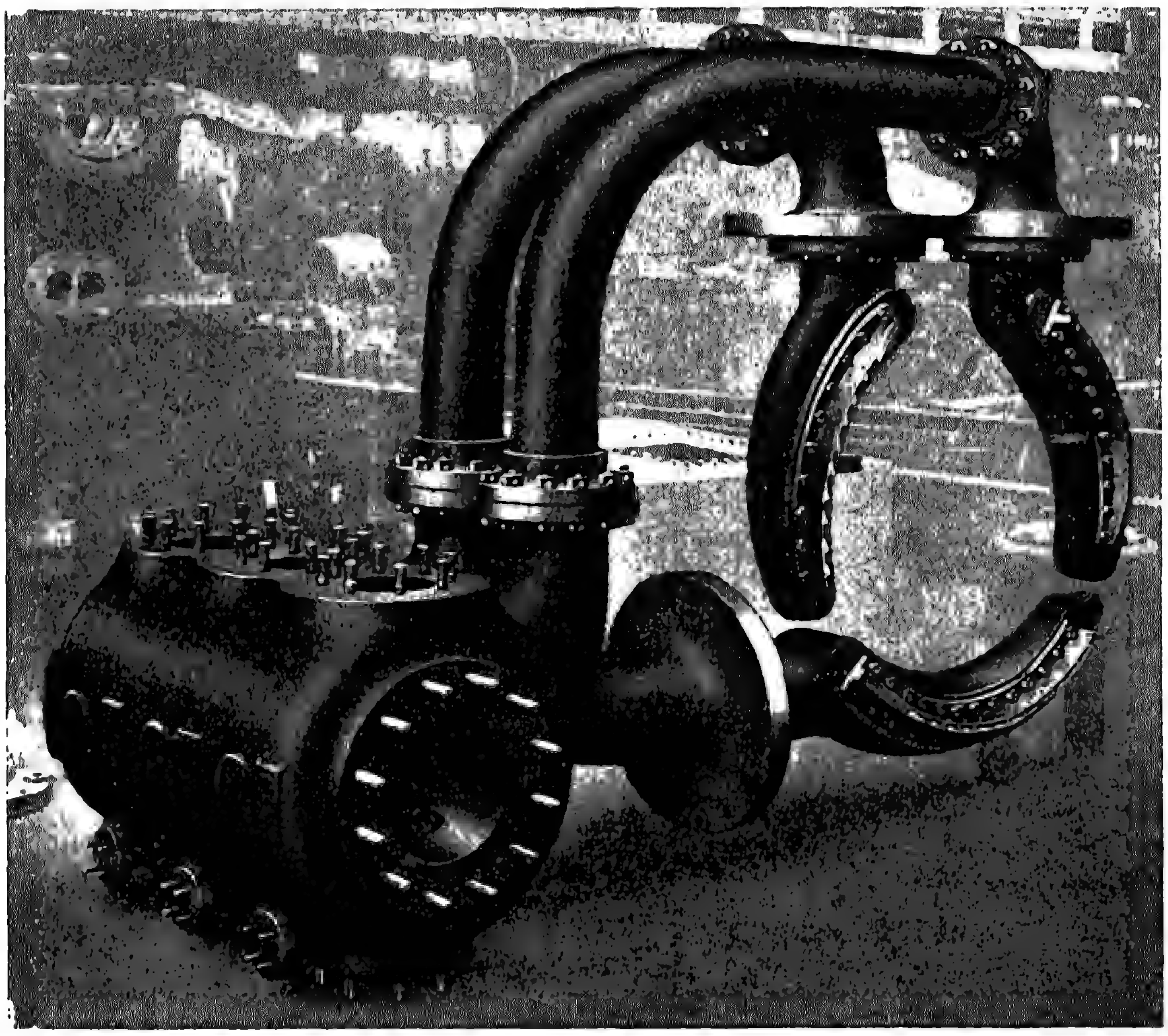


Fig. 19.—Nozzle Boxes and Connections to Steam Chest

pumped through an oil cooler of the surface-condenser type, and up into the bearings and oil-relay cylinder. A small hand-pump or steam-driven turbo-oil-pump is provided to lubricate the bearings when starting up the turbine.

THE METROPOLITAN-VICKERS COMPANY'S DESIGN

The steam turbines built by Messrs. The Metropolitan-Vickers Company are of the Rateau type generally, with a velocity wheel in the first stage. Their machines embody many points in design which are claimed to represent the outcome of the makers' extended experience in the production of large size units designed to operate under severe steam conditions.

Fig. 19 shows the three nozzle boxes and their connections to the steam chest. These are the only parts of the turbine subjected to high pressure and superheat. The boxes consist of steel castings, and are bolted at one end directly or through pipes to the steam chest, the other end being free to

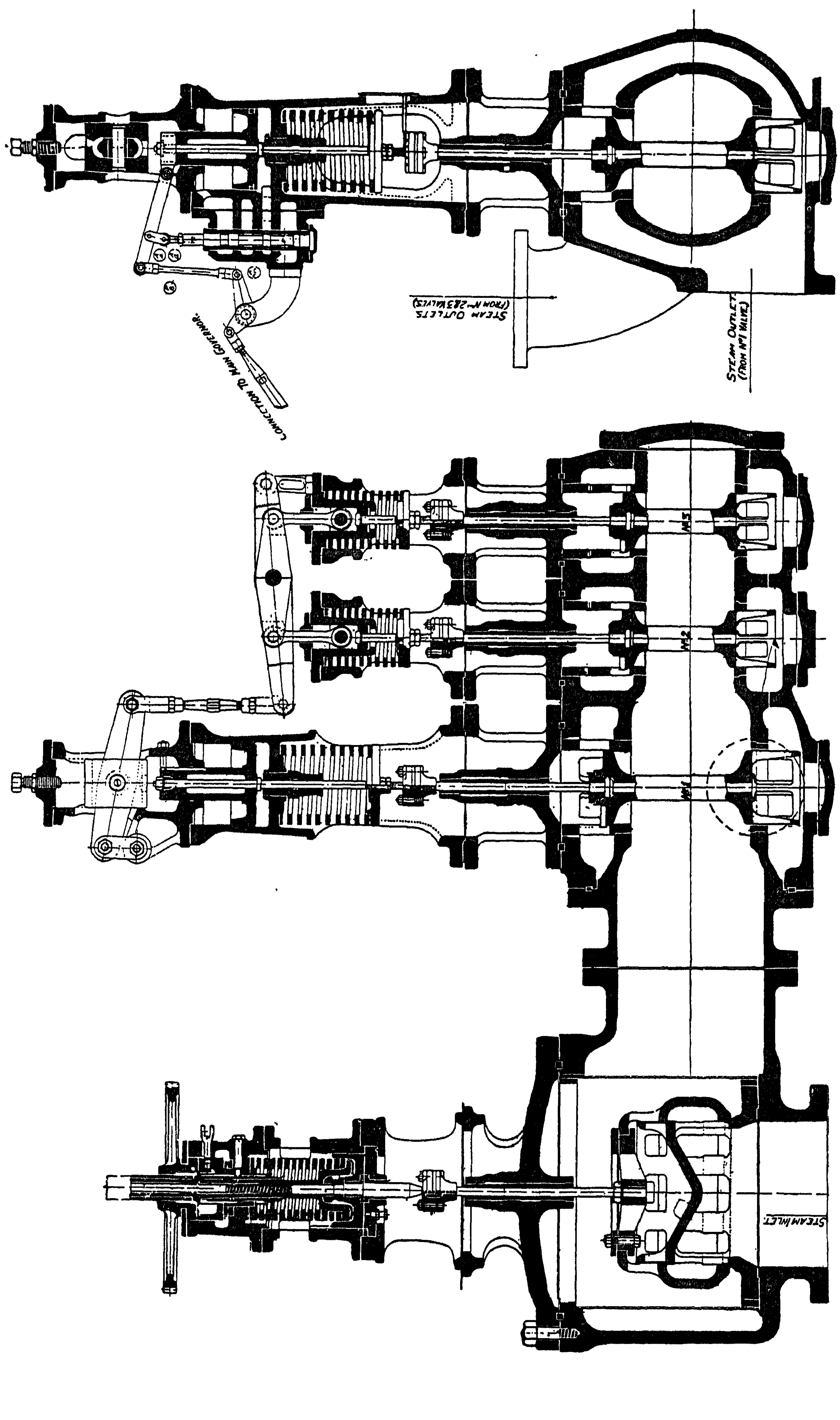


Fig. 20 —Sectional Drawing of Steam Chest for large Metropolitan-Vickers Steam Turbine

expand without stressing the casing in any way. The first set of nozzles admits steam up to a half load, the second up to full load, and the third comes into operation on overload.

Fig. 20 shows a sectional drawing of a steam chest as used on large turbines. The body of the steam chest consists of two pipes, an inner and an outer pipe. The inner pipe, to which steam is supplied from the main stop valve, is provided with three valves which admit the steam to three compartments formed within the steam belt between the inner and the outer pipes. Each of the compartments in turn communicates with its corresponding nozzle box and nozzles.

The valve operating mechanism is arranged so that the forces are acting in one plane, and the parts which are subject to wear are removed as far as possible from those parts which are subjected to high temperatures.

It will be seen that the three governor valves are actuated by means of oil pressure under the control of relay mechanism fitted to governor valve number one. As the load exceeds that which valve number one is designed to carry when fully opened, the spindle of this valve continues to move upwards, carrying with it a lever pivoted at one end and attached at the other end by means of a short vertical adjustable link to one end of a second lever, the other end of which is pivoted at a fixed point above the governor valve number three. This lever moves in the same plane as the lever situated above valve number one, and carries with it a floating lever which is connected at its extremes to governor valves two and three. The springs which close valves two and three are so adjusted that for an increasing load valve number three remains closed until valve number two is fully opened, and for a decreasing load valve two does not begin to close before valve three is fully closed. In like manner valve number one does not begin to close on a decrease in load until valve number two is fully closed.

Fig. 21 shows a section through a 12,500-Kw. turbine running at 3000 r.p.m., and embodies the makers' patent multiple exhaust blading in the low-pressure stages which they adopt in all large high-speed turbines. In the twelfth stage diaphragm the steam is divided into two parts by an annular division ring. The outer part of the diaphragm blades is designed to expand the steam to condenser pressure, whereas no appreciable fall of pressure occurs across the inner part of the blades.

The corresponding moving blades are again subdivided by a ring, and whereas the outer part of the blades is shaped to utilize the kinetic energy acquired by the steam in the outer part of the preceding diaphragm, the inner part is so shaped as to allow the steam to pass without appreciable expansion. This steam quantity then enters the next stage which is formed on similar lines, and the steam passing through the inner part of this stage is finally expanded to condenser pressure in the last stage whose area is available for the purpose. The number of stages embodying multiple-exhaust blading is varied according to the output speed and exhaust pressure. The small additional leakage losses encountered due to this system are claimed to be of small moment in comparison to the advantage accruing from the use

of smaller diameter discs and shorter blade lengths, the effective blade area of the last expansion being the sum of the outer areas of the multiple stages plus that of the complete last stage.

The nest of tubes which is shown in the lower part of the turbine casing, and which passes across the full length of the casing, represents a feed heater forming part of the company's patent feed-heating system which is embodied in all their medium size and large turbines.

The feed-heater installation consists of a main heater of the surface type, fitted with the turbine casing across which it extends, and a subsidiary surface heater which is arranged in a horizontal position at right angles to the main heater. The subsidiary heater is arranged vertically merely for showing clearly all the connections in diagrammatic form (fig. 22). The method of operation is as follows:

The condensate is withdrawn from the condenser by the extraction pump C, which pumps the feed, first through the lower portion D of the subsidiary heater B, second through the main heater A, and then through the upper portion of the subsidiary heater to the feed outlet F.

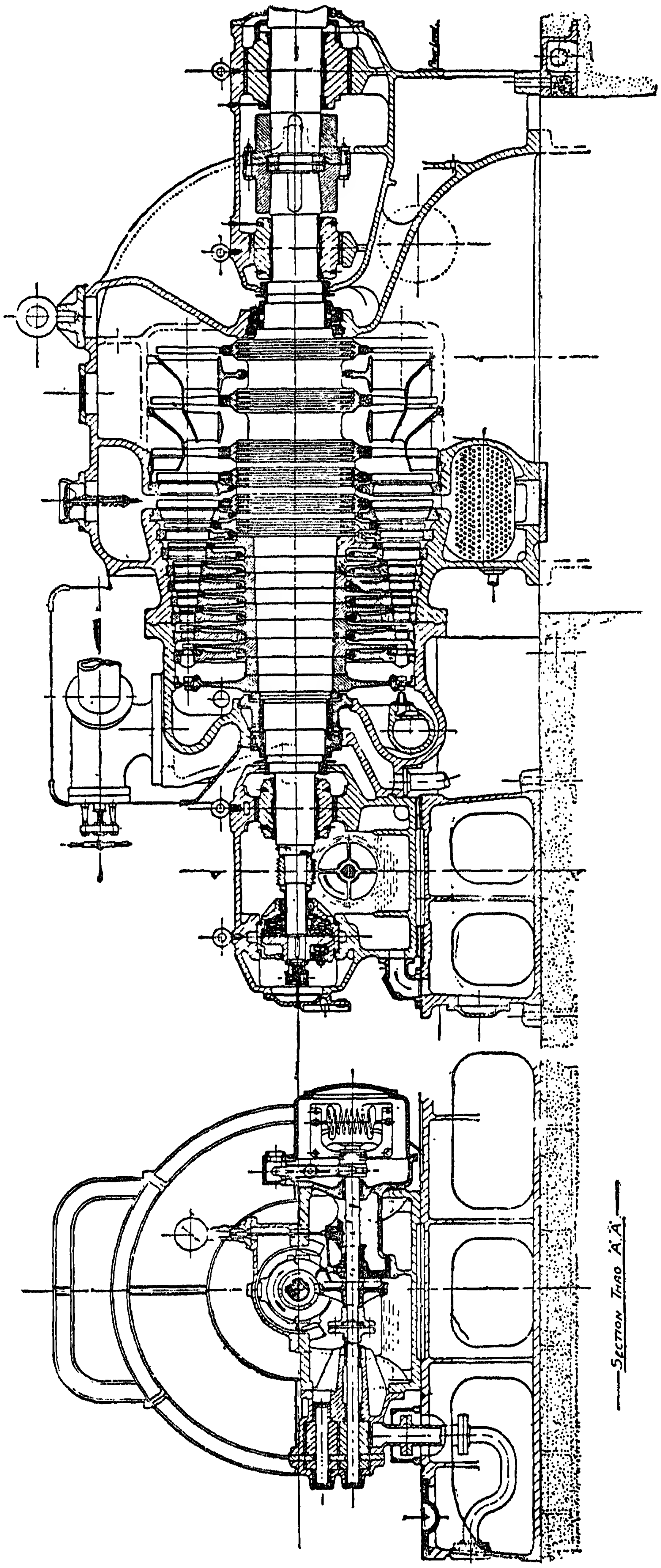


Fig. 21.—Section through a 12 500-Kw.-3000-r.p.m. Metropolitan-Vickers Steam Turbine

The tubes of the main heater are exposed to steam, which is diverted

from the turbine at a pressure corresponding to a vacuum of about 21 in. The upper portion of the subsidiary heater utilizes leakage steam, entering at G, from the high-pressure turbine glands.

The condensate from the steam condensed in the main heater is passed into the lower portion of the subsidiary heater, where it comes into contact with the tubes through which the feed water passes on its way to the main

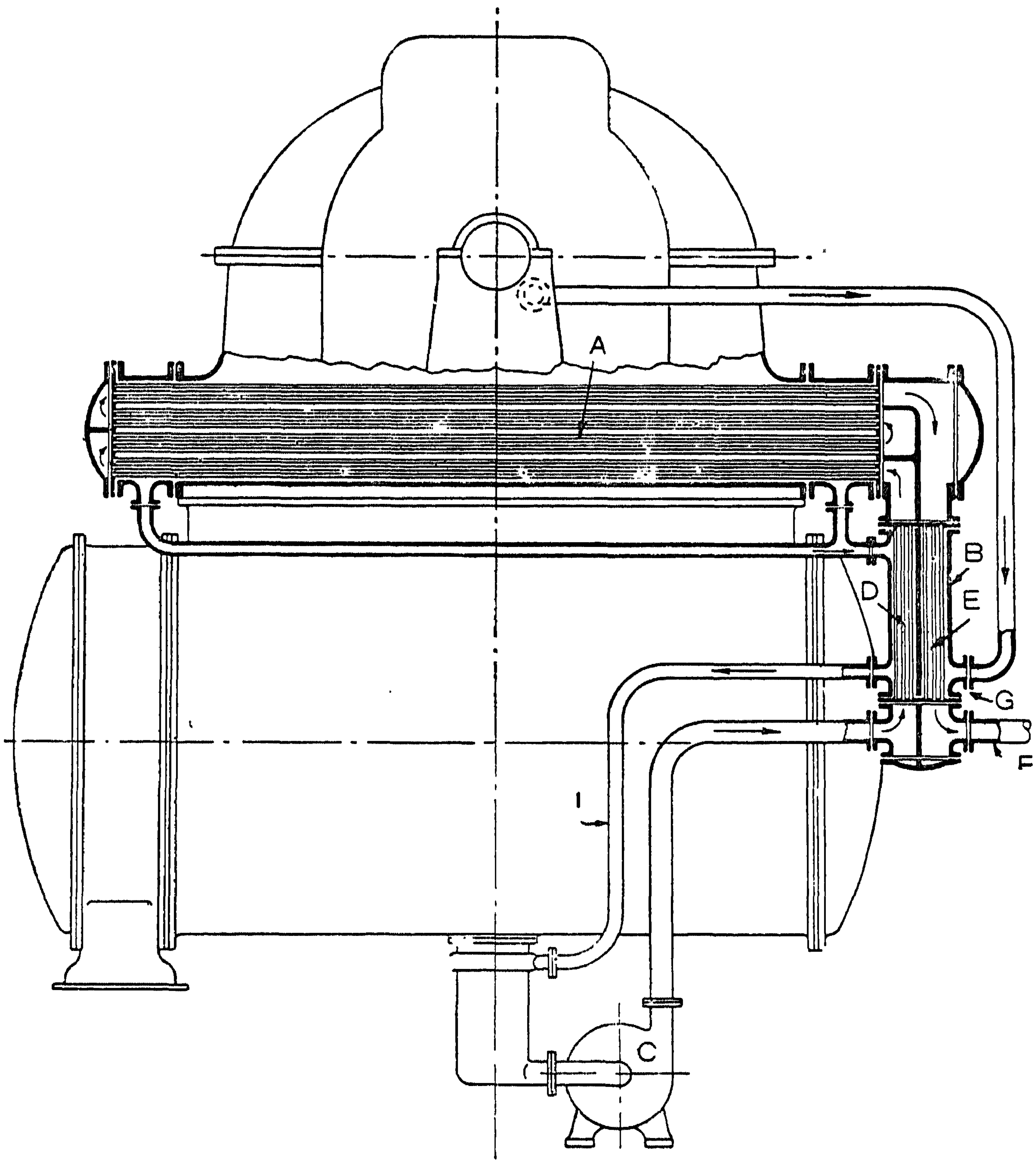


Fig. 22.—Arrangement of Feed Heater on 18,750-Kw. Metropolitan-Vickers Turbine Alternator

heater. The condensate from the steam condensed in the upper portion of the subsidiary heater drains into the lower portion, from which the whole of the condensate is withdrawn by means of a pipe connection to the main condenser.

Numerous test results have shown a gain in fuel consumption of upwards of 5 per cent due to the use of this system.

The blade system generally follows standard Rateau practice, except that the low-pressure blades are axially tapered to reduce the centrifugal stresses.

The Metropolitan-Vickers Company use solid couplings between turbines and alternators, but where gear drives are used a flexible coupling is usually employed between the turbine and the gear pinion.

CHAPTER VII

The Zoelly Type Turbine

The Zoelly turbine as built by the original makers, Messrs. Escher, Wyss, & Co., of Zurich, can be taken as a representative pure multi-pressure stage turbine, which in its general principles has held a leading position in all countries.

A number of British manufacturers have adopted this type under licence, and the following illustrations refer to machines of the Zoelly type as built by Messrs. The English Electric Company. The designs shown differ in some important respects from Messrs. Escher, Wyss, & Co.'s practice.

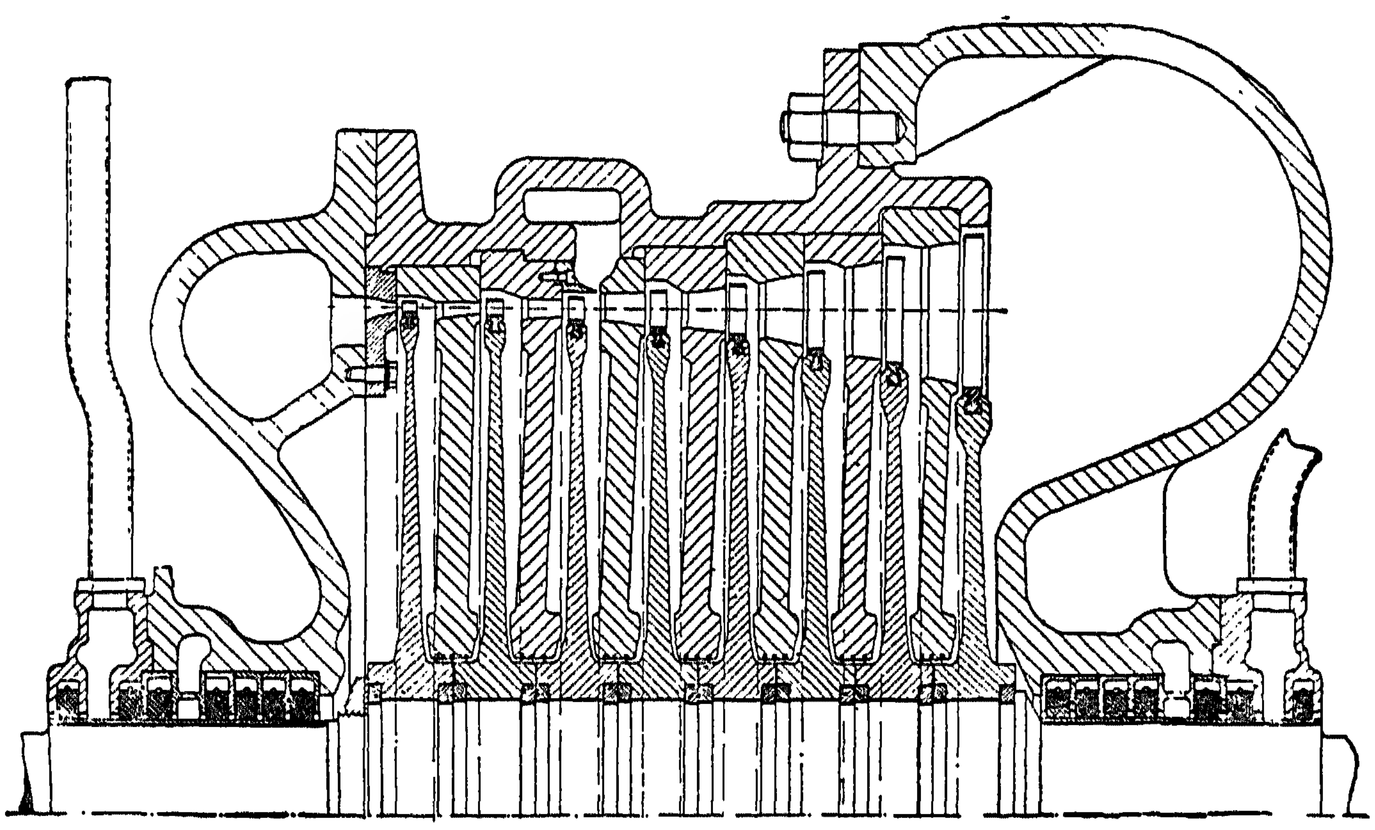


Fig. 23.—Simple Impulse Type Zoelly Turbine

Fig. 23 illustrates a Zoelly turbine where the pure pressure stage principle is retained. The question of output, steam, and speed conditions determine what the form of the first stage shall be.

The rotor is composed of a high-quality steel shaft on which are mounted the steel discs carrying the impulse buckets or blading. These discs are separated from one another by a series of diaphragms carrying the stationary blades or nozzles. These diaphragms divide the casing into pressure stages.

The high-pressure end of the casing carrying the steam belt and first nozzles is made of cast steel, as the high pressures and temperatures occur only in this part of the machine.

Steam-sealed glands are fitted between the casing and shaft at either end.

The rotor is supported on pedestal bearings with pressure-fed lubrication. The axial adjustment between the wheels and diaphragms is determined by the thrust block mounted on the end of the shaft. The exhaust-end pedestal is made large enough to take the bearing of the driven unit.

The admission of the steam is controlled by an oil-relay governor which uses the same oil as the main bearings.

An overspeed device actuating an automatic emergency governor is fitted to shut down the machine should the speed pass a limit, which is predetermined.

The emergency governor closes both throttle and stop valve. An auxiliary automatic oil-pump is fitted which comes into use when the machine shuts down or starts up. This is cut out when the main oil-pump, driven by worm from the main shaft, comes into action.

The low-pressure portion of the casing is of cast iron suitable for moderate pressures and temperatures.

The casing is divided on the horizontal line, and the top half may be lifted without breaking any pipe joints or dismantling the nozzle or governor gear. Supporting feet at each side of the casing are arranged at the low-pressure end, and are cast in with the exhaust chamber. These feet are

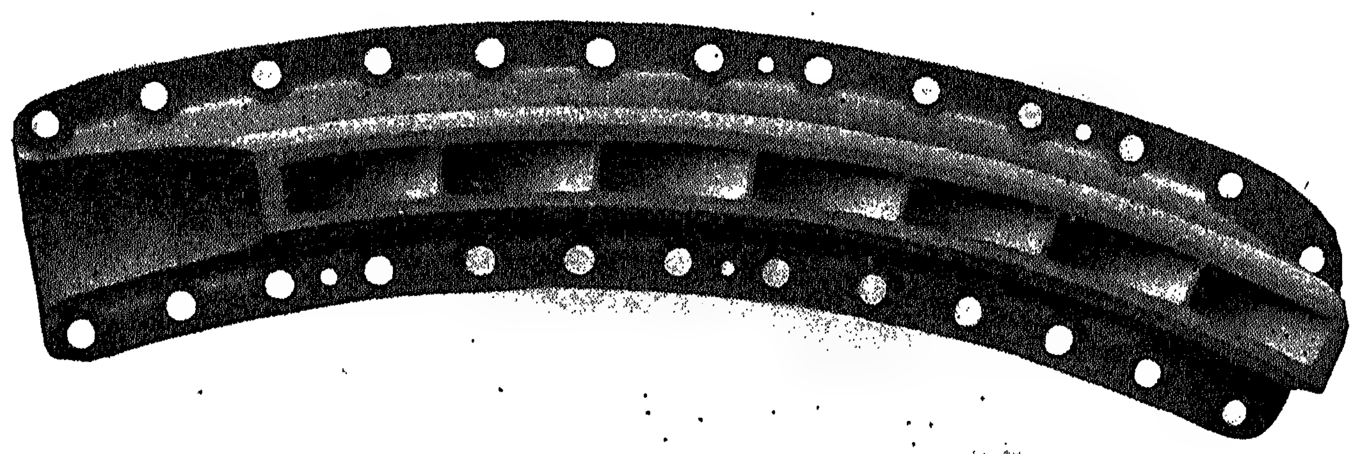


Fig. 24.—H.P. Nozzle Block

free to slide along the bedplate to give the necessary movement when expansion takes place.

The turbine casing is anchored at the exhaust-end pedestal bearing so that expansion takes place towards the high-pressure end of the machine.

The steam-end bearing pedestal which carries the thrust block is connected to the casing. It can slide freely over a steel feather in the bedplate in an axial direction, and thus follows the expansion of the casing. At the same time the thrust block automatically corrects the differential expansion which takes place between the rotor and casing.

The first set of nozzles is carried in castings which are bolted to the cast steel cover at the high-pressure end of the turbine. The steam passages are cast into the nozzle plates, and the nozzle exits are machined.

Fig. 24 shows one of the nozzle plates for the first stage. They do not generally extend round the whole periphery in the first stage, two or more sections being used as required for the quantity of steam to be passed. The nozzles in the later stages are similarly formed in the diaphragm plates.

The centre lines of the nozzle are not tangential to the mean diameter of exit, but are slightly inclined inwards so that the steam jet shall strike the blades at the correct angle.

The diaphragms are of cast iron or steel, and are dished slightly to withstand the varying pressure during rapid changes of load without buckling.

The diaphragms are made in halves with horizontal joints, the two halves being joined with bronze keys. The top halves are held in the turbine cover with locking screws.

Labyrinth glands are used on the diaphragms between their centres and the shaft. These are made of soft copper strips of knife-edge section fitting into parallel grooves so that the knife edges point inwards and practically touch the shaft. These are shown in fig. 25, and in the case of touching the soft copper is merely pressed into a running fit on the shaft.

The shaft is made from a solid steel forging annealed and reannealed between the varying stages of manufacture. It is finally ground to gauge.

The discs or wheels are solid steel forgings, and are repeatedly annealed during machining.

The surfaces are polished to reduce frictional losses.

The discs are profile section, and are mounted on the shaft with supporting rings for which the shaft is grooved. The discs and rings are then keyed into position.

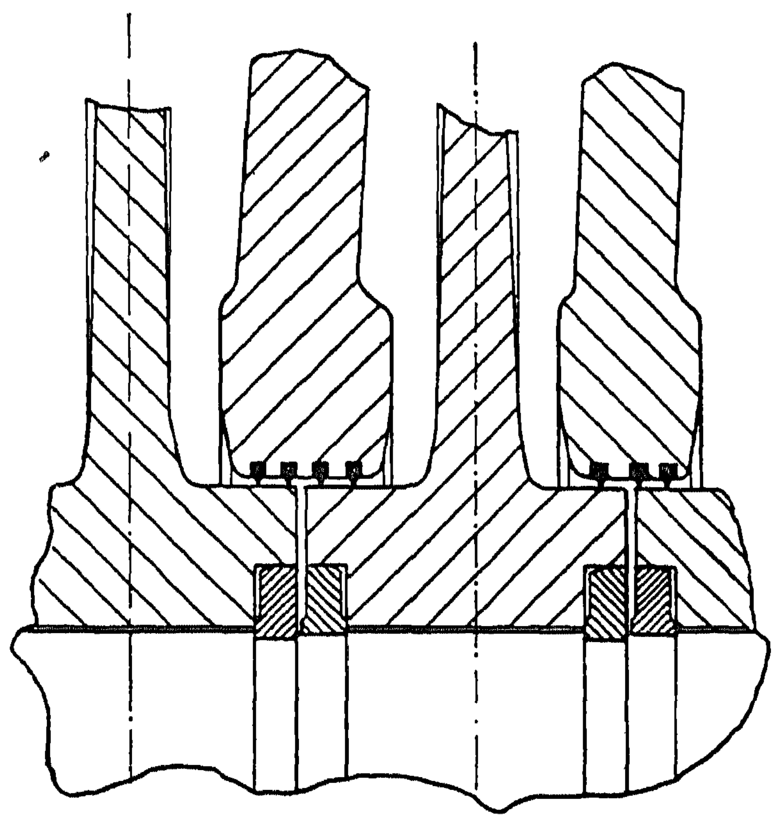


Fig. 25.—Diaphragm Glands and Wheel Fixing

Fig. 25 shows clearly the method of fixing. The wheel does not actually touch the shaft, and can be mounted and dismounted regularly; moreover, there is no danger of damage to shaft surfaces in dismantling. The points of support are at the outer edges of the hub, and the centre of the disc, where the maximum stretching due to rotation occurs, is not in contact with the shaft or supporting ring. When rotating at high speed the stretch at the centre tends to close the outer edge of the hub, producing a tighter fixing and not tending to loosen the disc on the shaft when fully stressed.

The discs with their blading are balanced singly before erection, and the rotor complete with all discs is balanced finally.

The moving blades are made from nickel-steel bars, machined to gauge size, and the surface polished.

The blade roots of T section fit into corresponding grooves in the peripheries of the wheels. The blades are inserted through windows which are afterwards closed by special wedge-section locking pieces. The full width is given at the blade root.

The spacing of the blades is effected by spacing pieces which are generally an integral part of the blades, but occasionally made separately.

The shrouding is a steel strip, and is fixed by machined tangs on each blade end. The tangs fit into rectangular holes in the shrouding and are riveted over.

The longer blades in the low-pressure end taper to the tips.

The glands at both high- and low-pressure ends of the turbine are of the

carbon-ring type. They are shown in fig. 26. They consist of a number of carbon rings held close round the shaft by means of springs. Spaces are left between pairs of rings into which steam at atmospheric pressure is led. This forms an effective seal.

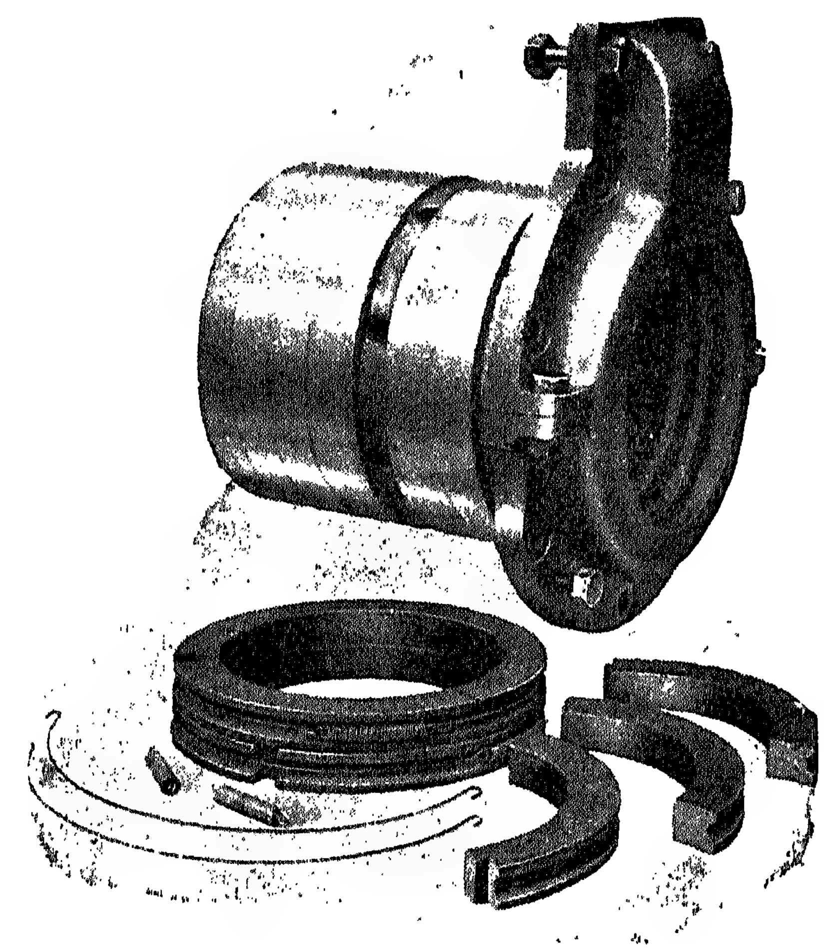


Fig. 26.—Standard Shaft Gland

The rings are made in three segments of graphitic carbon, and when assembled surround a sleeve fixed on the shaft.

The thrust block is of the Michell self-lubricating type.

The governor is of the vertical totally-enclosed centrifugal type, and is designed to give fine regulation. The governor weights are held in position on the vertical arms of bell-crank levers by compression springs. The pressure oil supply is used for lubricating all parts of the governor.

The motion of the governor sleeve, which is free to move on the vertical governor spindle, is communicated to a floating lever, which in turn operates

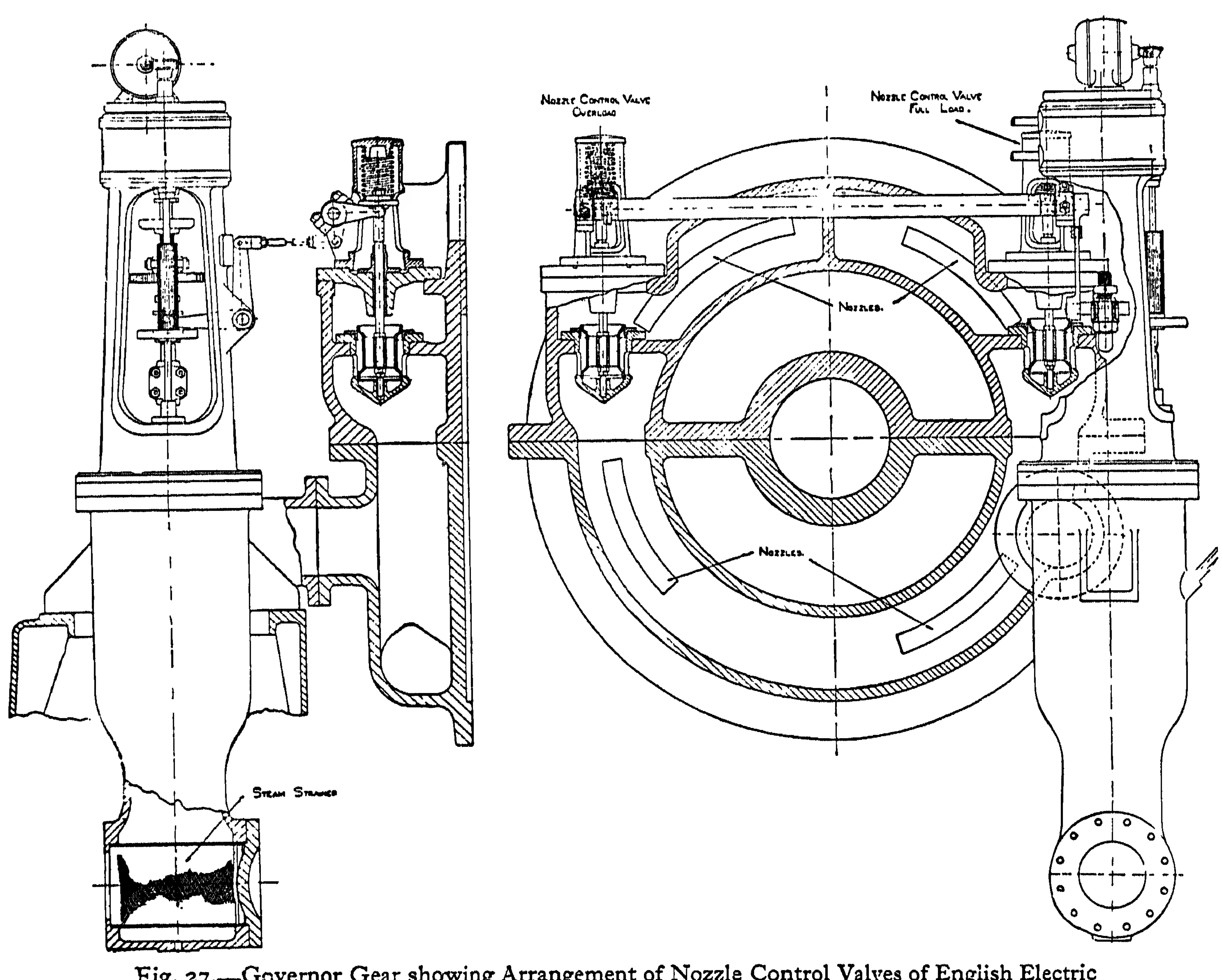


Fig. 27.—Governor Gear showing Arrangement of Nozzle Control Valves of English Electric Company's Zoelly Turbine

a pilot valve admitting oil under pressure to the top or bottom of the oil-relay cylinder. A drop in the speed of the turbine causes a downward motion of the governor sleeve. This alters the position of the pilot valve, and oil flows under the relay piston, lifting it, and through it the main throttle valve, until the turbine speed comes up to the normal. The reverse occurs if the turbine speeds up.

The nozzle control valves are shown in fig. 27. The high-pressure end of the turbine where the steam enters is separated into three sections by means of auxiliary nozzle control valves. These are directly actuated by the relay piston of the governing gear.

The first portion of the travel of the oil-relay piston affects the steam supply to the first section of nozzles, but movement beyond this opens the valves consecutively for full load and overload.

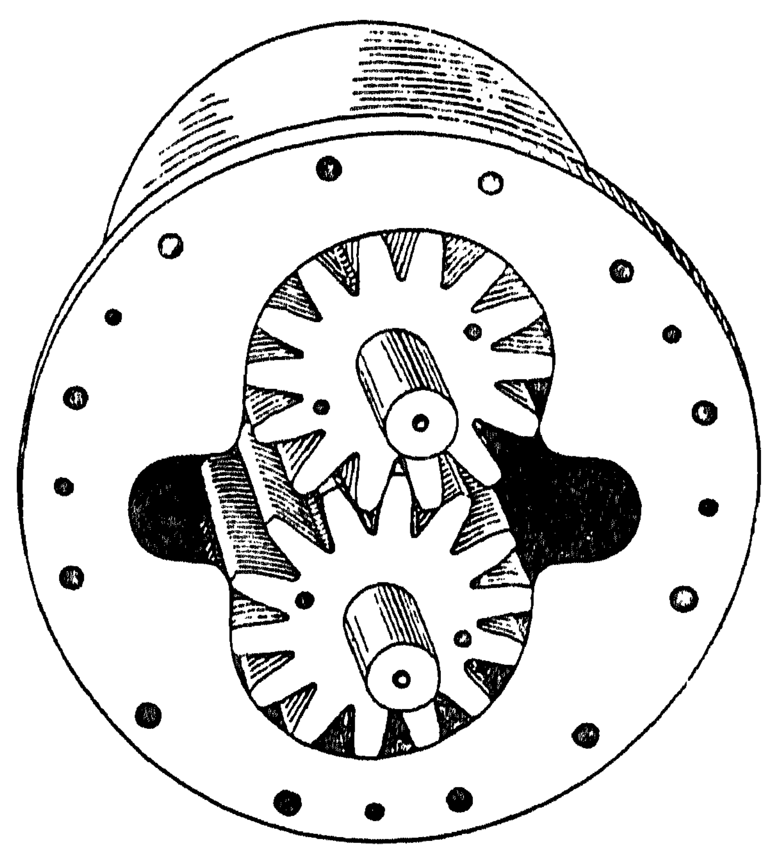


Fig. 28.—Oil-pump

The emergency governor and overspeed device consists of an unbalanced ring placed eccentrically on the shaft next to the thrust block, and held in position central to the shaft by a compression spring. The compression of the spring is overcome at the predetermined maximum speed, and the ring, due to the unbalanced centrifugal force, becomes eccentric to the shaft and makes contact with a trigger which trips the valve-operating gear. This closes simultaneously the combined emergency valve, stop valve, and the main governor valve.

The trigger can be tripped by hand so that it can be seen that it is in working order at all times.

The oil for the lubrication of the bearings and thrust block and for the operation of the governor gear is supplied from a rotary pump shown in fig. 28. This pump is driven by worm gearing from the turbine shaft. The pump, as can be seen, has no valves.

Leaving the pump, the oil first passes through the governor gear and through a cooler to the bearings and thrust block, and then, flowing by gravity, it passes to the main reservoir through a strainer. The strainer is so arranged that plates can be withdrawn for cleaning whilst running. An auxiliary pump of the duplex reciprocating type is provided to guard against failure. It is brought into operation by any failure in the oil pressure.

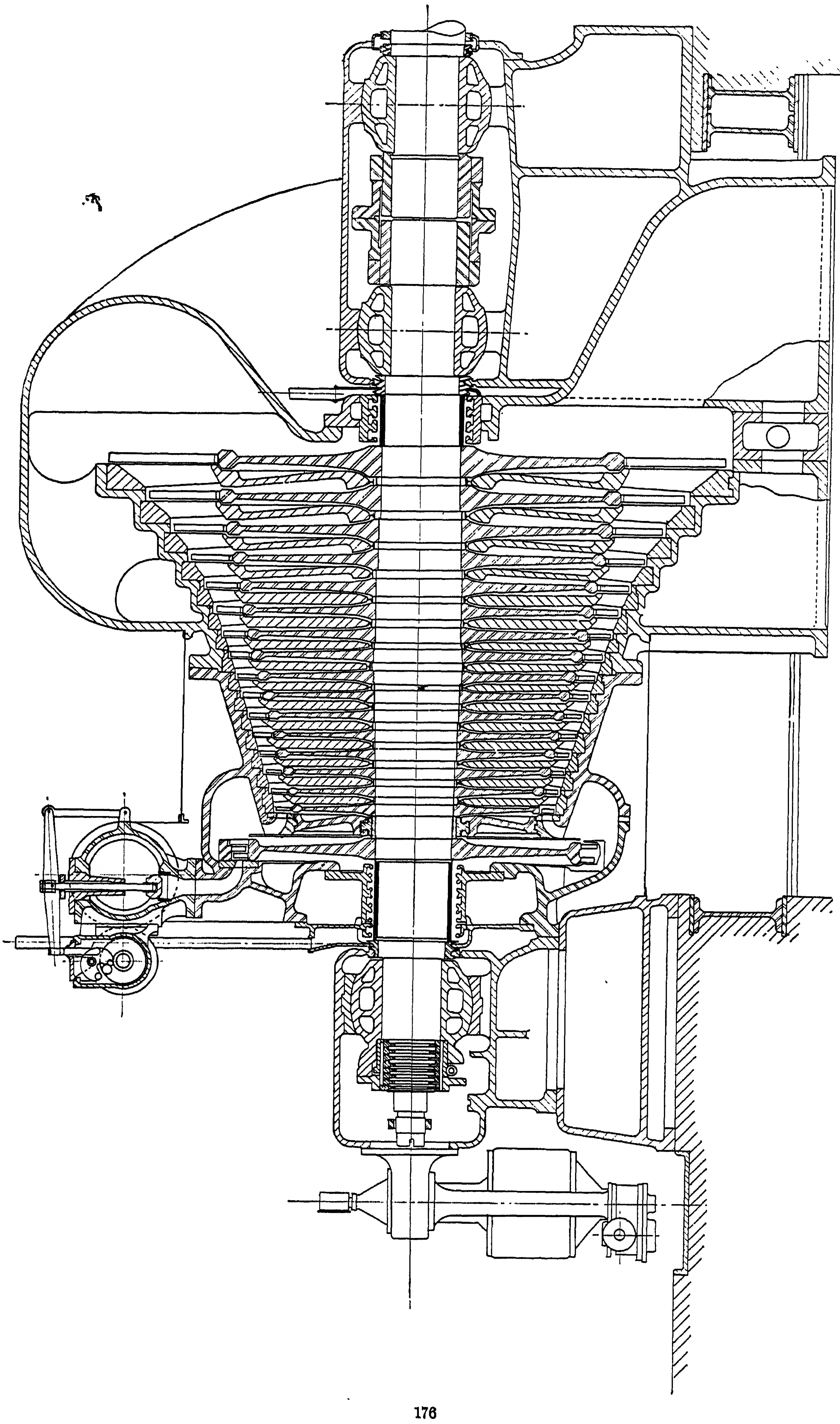


Fig. 29.—15,000-Kw. Turbine

CHAPTER VIII

Curtis Turbine

The British Thomson-Houston Co., Ltd., of Rugby, builds turbines on principles similar to those built by their parent company, the General Electric Company of America, the original manufacturers of the Curtis turbine. Their earlier designs consisted of two or more pressure stages,

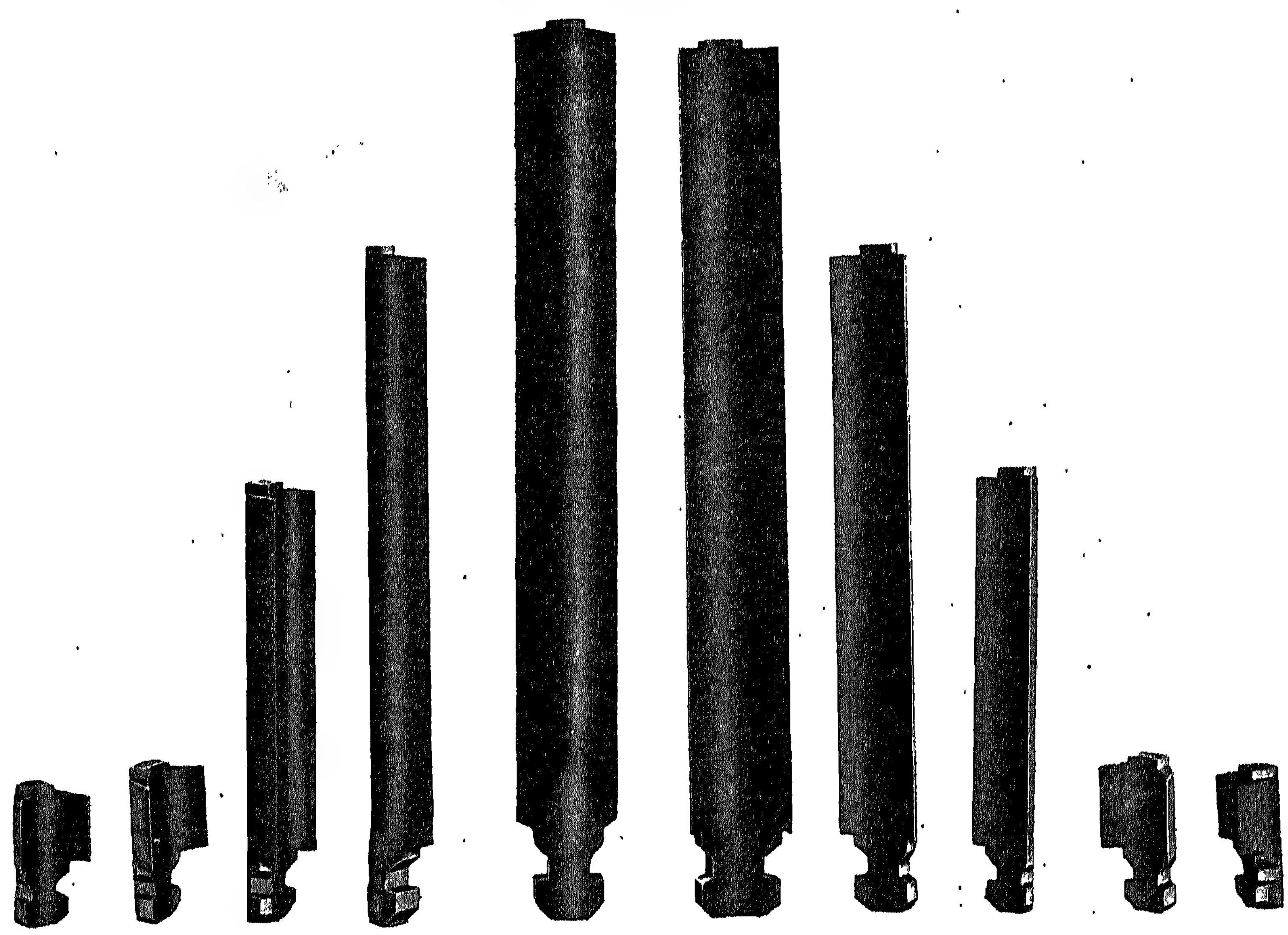


Fig. 30.—Moving Blades for Curtis Turbines

each compounded for velocity, but later designs show one stage compounded for velocity, and the remainder as single-pressure stages.

Fig. 29 illustrates a 15,000-Kw. turbine designed to run at 1500 r.p.m. The same firm has built a 30,000-Kw. machine on similar lines, except that this consists of one velocity stage and fourteen pressure stages, against the sixteen stages shown for the 15,000-Kw. design. It will be seen that the disc diameters increase with succeeding stages with a view to obtaining an increase in blade area and blade speed corresponding as nearly as possible to the ideal conditions demanded by the expanding steam. The design is characterized by the retention of the single-flow principle at the low-pressure stages, notwithstanding the large blade area needed.

The moving blades are usually of phosphor bronze, the shorter blades being cut from bars cold drawn to finished section. Longer blades are milled from heavier section bars, the roots being left at the full section to give greater strength; the blades then forming their own distance pieces.

Where the blade stresses are too high for convenient use of phosphor bronze, steel blades are adopted, milled out of the solid bar. The blade roots are either dovetailed (fig. 30) and fit into corresponding slots in the

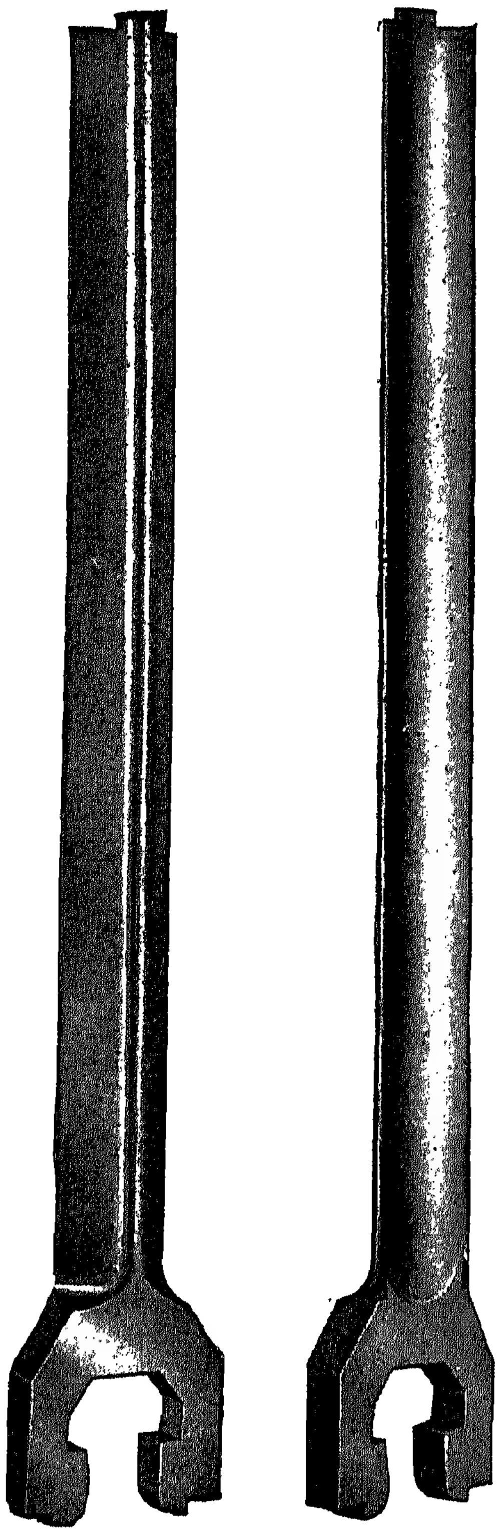


Fig. 31.—Moving Blades with Inverted Dovetails, for Curtis Turbines.

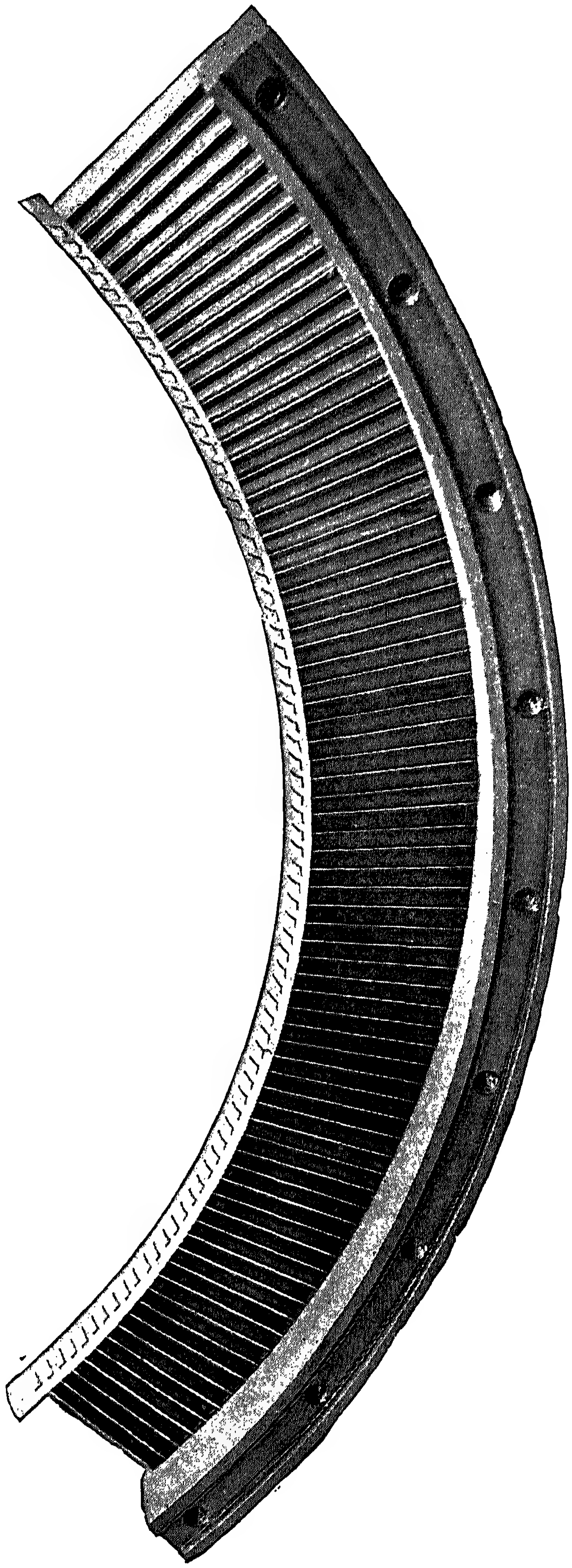


Fig. 32.—Section of Guide Blades

rim of the disc, or where the disc stresses are such as to make it preferable to reduce the rim thickness to a minimum, then the blade roots are milled with an inverted dovetail as shown in fig. 31.

Fig. 32 shows a section of guide blades to reverse the direction of motion of the steam between velocity stages. These blades are made in a similar manner to the running blades, and are fitted into a cast-iron ring which is bolted to the casing or the nozzle plate. The arc covered by the guide blades is practically equal to that covered by the first stage nozzles, so that where the nozzles are confined to the top half of the casing the same applies to the guide-blade ring.

The British Thomson-Houston Company adopt flexible claw-type coup-

lings between the turbine and the driven machine. These couplings are contained within the centre bearing housing, and are well lubricated with oil from the bearings, special ducts being provided for the purpose. Fig. 33 shows a coupling of this type designed for a comparatively small power machine closed. The coupling bolts need to be dimensioned to transmit the full torque in shear. Fig. 34 shows four parts of a similar coupling designed for a 10,000-h.p. turbine. The barring gear engages in the teeth that can be seen on the outside of the coupling flanges.

The turbine discs are pressed on to the shaft over bronze rings to avoid any danger of tearing the shaft surface in forcing the discs on or off.

The governor is of the centrifugal type carried on a vertical shaft which is driven by a worm and worm wheel from an extension of the turbine main shaft.

The governor does not actuate the controlling valves directly, but controls

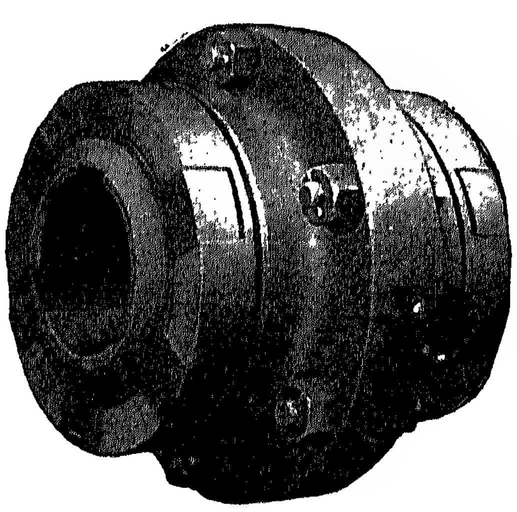


Fig. 33.—Claw-type Flexible Coupling

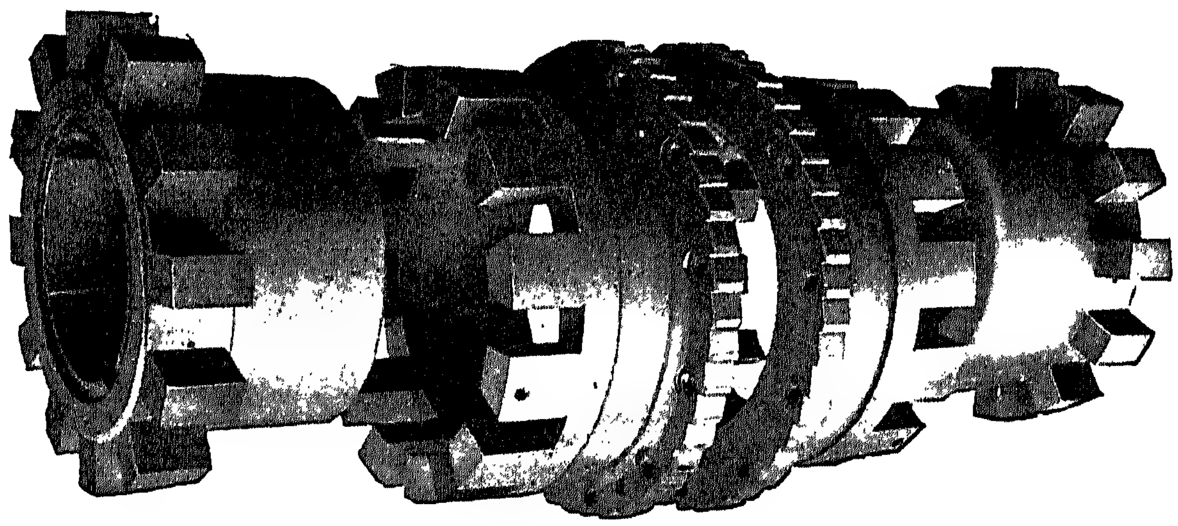


Fig. 34.—Claw-type Flexible Coupling to Transmit 10,000 h.p.

a small balanced pilot valve (fig. 35), which in turn controls the supply of oil under pressure of about 50 lb. per square inch gauge to the servo-motor, by means of which the controlling valves are hydraulically operated. The piston of the servo-motor is attached to the end of the cam-shaft as shown in fig. 35. The movements of the governor are transmitted to a pilot valve through a floating lever, to which the pilot valve spindle is attached. One end of the floating lever is pivoted to the governor lever and the other end to the rack, which engages with a pinion on the servo-motor spindle or cam-shaft. Thus when the governor lever rises or falls the pilot valve is moved up or down, and so admits oil to one side or other of the servo-motor piston. This causes the piston to rotate, which in turn rotates the cam-shaft and so opens or closes the necessary controlling valves.

When the servo-motor piston rotates, the pinion on the end of the cam-shaft also rotates, and moves the rack up or down. This restores the pilot valve to its original position. Therefore the correct relation between the length of the steam belt and the load conditions is always maintained, and the speed of the turbine kept practically constant and free from hunting.

To attain stability the governor is adjusted to give a small drop in speed with increase in load, this drop being usually $2\frac{1}{2}$ per cent between no load and full load.

To facilitate synchronizing and the transfer of load from one machine to another, the governor is provided with a hand-operated synchronizing arrangement, by means of which the speed can be varied 5 per cent above or below normal speed when running.

Steam is admitted to the turbine through a number of fixed nozzles usually bolted to the controlling valve chest. The nozzles are divided into groups, and the admission of steam to the groups of nozzles is controlled

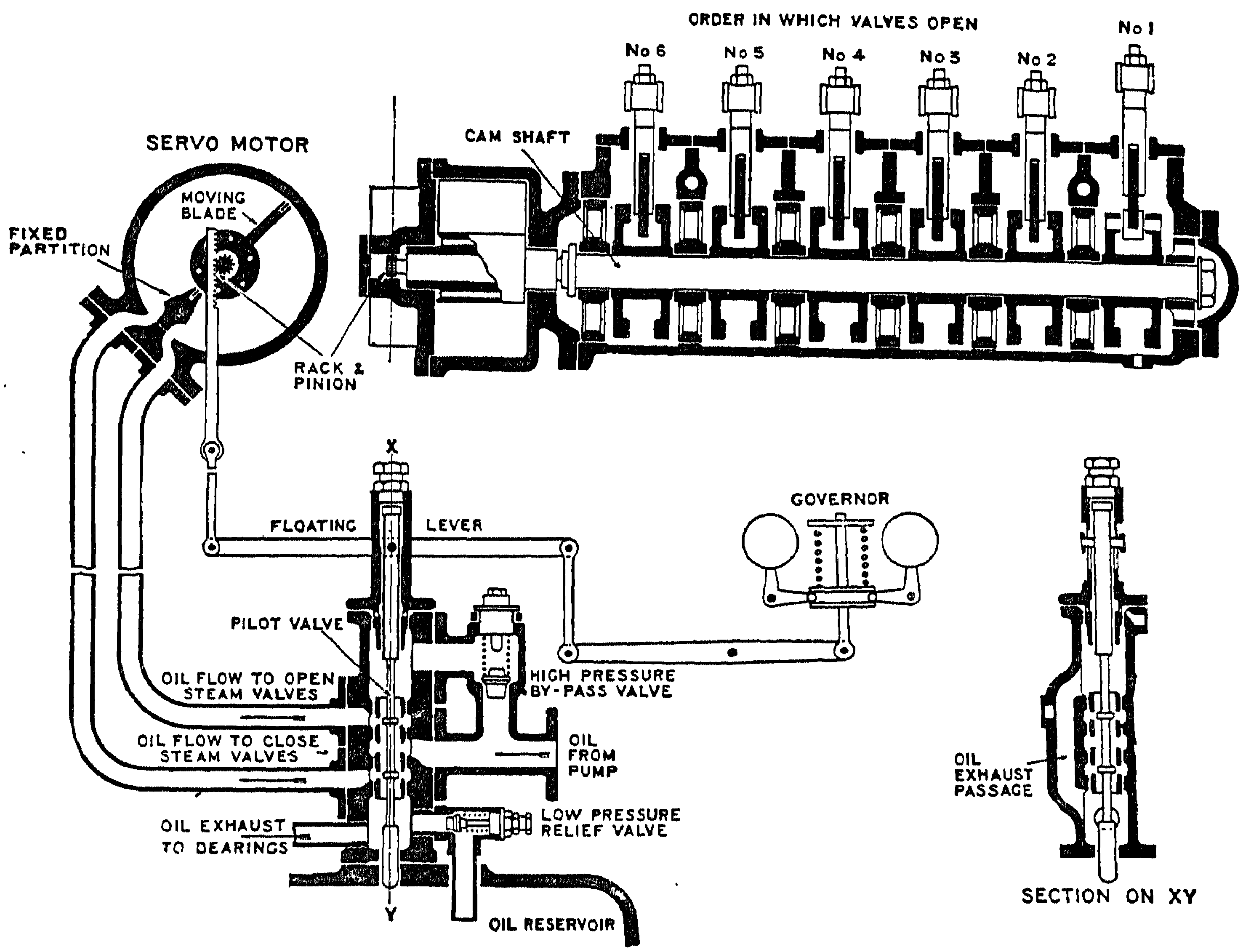


Fig. 35.—Governor and Nozzle Control Gear for Curtis Turbine

by a series of poppet valves. The spindle of each valve extends through a stuffing box, and is secured to a lever, one end of which is pivoted to a link attached to the valve chest, whilst the other end is connected to a cam mechanism, by means of which the lever is operated, thus causing the controlling valve to open or shut as required.

Turbines of over 2000-h.p. capacity are provided with a turbine-driven auxiliary pump to flood the bearings when running the turbine up and shutting down, and to come into operation in case of failure of the main oil supply. In the case of smaller turbines, each bearing is provided with lubricating rings which take the place of the auxiliary oil-pump, and an ample oil-well is arranged to feed the rings.

CHAPTER IX

Impulse-reaction Type Turbines

The impulse-reaction type of turbine, with the adoption of the single-impulse stage compounded for velocity in place of a series of reaction stages at the high-pressure end of the turbine, results in a substantial shortening of the turbine, so that it has been found possible to build such machines in a single casing for large outputs at high speeds, with the critical speed 50 per cent above the normal speed.

It is further claimed that the lower blade efficiency in the impulse stage is compensated for by the reduced leakage losses, so that the over-all efficiency of the stage is no less than that of the series of reaction stages which it replaces.

The adoption of the initial impulse stage confines the steam at high pressure and temperature to the throttle valves and nozzle box and plate, so that for normal steam conditions these parts are the only parts that require to be made of cast steel.

Fig. 36 shows a section through a 20,000-Kw. impulse-reaction turbine built by Messrs. Richardsons, Westgarth, & Co., Ltd., of Hartlepool. In a turbine of this description there is, of course, under certain conditions of working, a considerable end thrust, but this is compensated for by means of a single dummy piston at the high-pressure end so arranged that the steam passing the dummy is expanded to the pressure prevailing at the inlet of the low-pressure section of the turbine, so that the loss in heat drop of the balancing steam is only that due to the drop over the intermediate-pressure section.

A single dummy does not fully balance the whole of the end thrust under every condition, and a thrust block of the Michell type is fitted to take up any unbalanced end thrust and for the purpose of adjusting the alignment.

In the larger units of 10,000 Kw. and above, two steam inlet valves are fitted which enable smaller valves to be used, and to ensure more equal temperature distribution over the high-pressure end of the cylinder.

A considerable amount of attention has been given to the design of a stream-line type of exhaust bend to minimize the losses between the last row of blades in the condenser inlet.

In order to reduce the depth of the condenser pits in large units, the bedplate supporting the exhaust end of the turbine has been designed to form two girders sufficiently deep to carry the weight of the turbine and the condenser, and to take the place of girders between the two supporting concrete blocks.

The shafts of smaller machines are of simple drum construction with shaft ends at either end, the disc carrying the impulse blading being bolted to the high-pressure end of the drum. In the present design it will be seen that the rotor is of built-up construction, a shaft of comparatively small

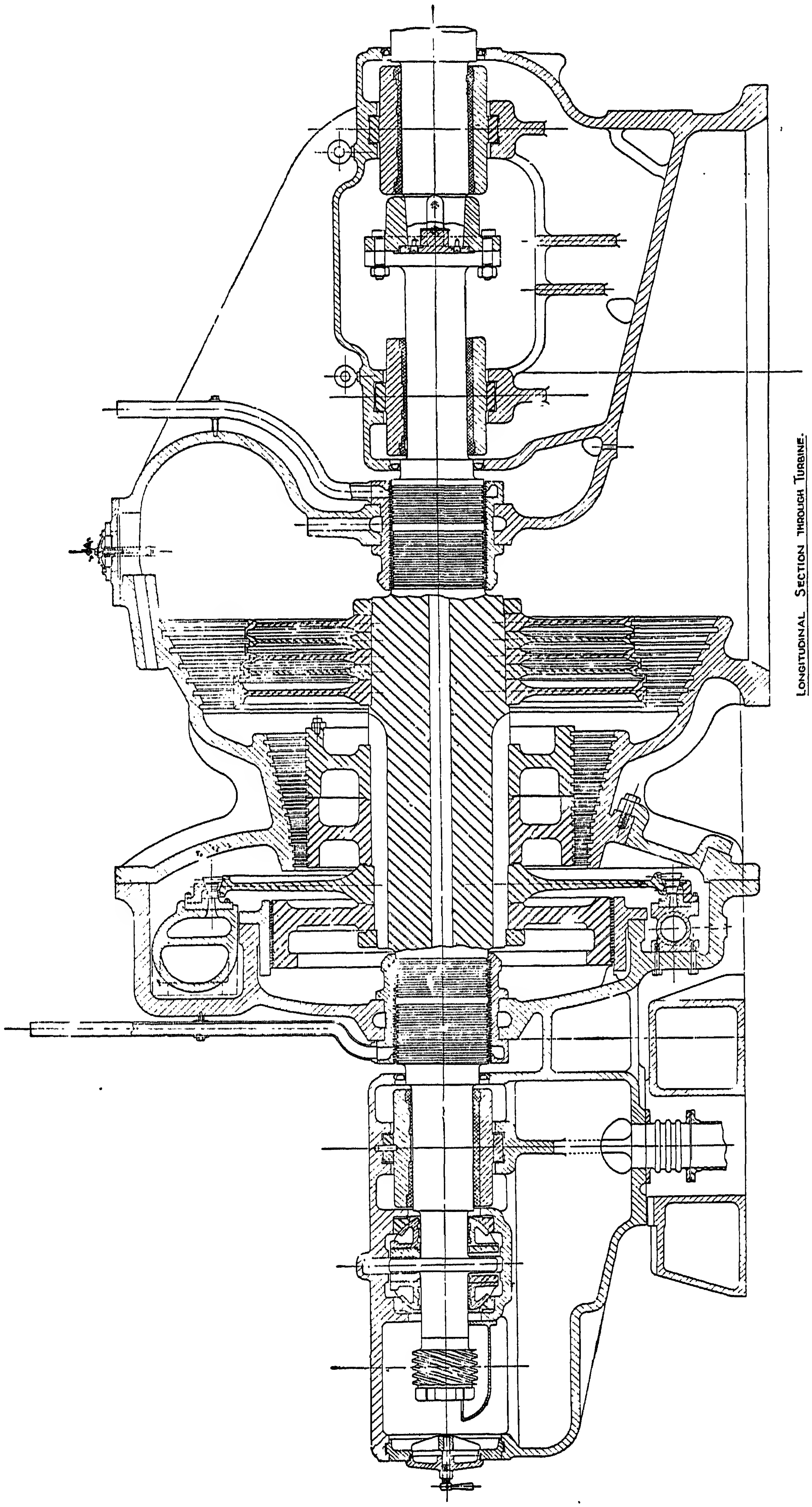


Fig. 36.—20,000-Kw., 1500-r.p.m. Impulse-reaction Turbine by Messrs. Richardsons, Westgarth, & Co., Ltd., Hartlepool

maximum diameter running through from end to end. The intermediate pressure reaction blading is carried in two hollow drums shrunk and keyed on to the shaft. The low-pressure blading is carried on a series of discs whose rims butt against each other, and whose hubs are shrunk and keyed to the shaft, a nut at the end of the discs locking the whole. It will be seen

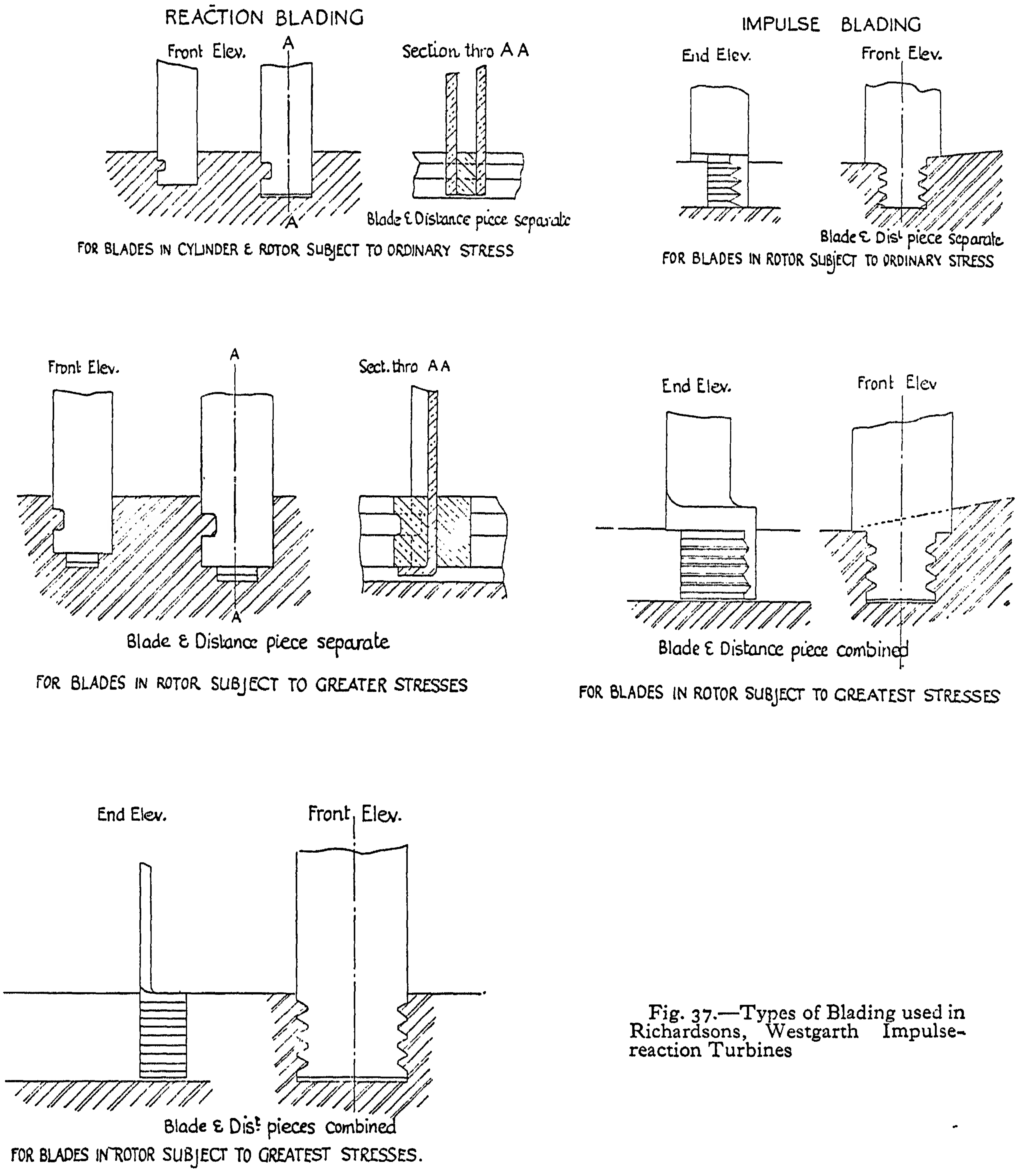


Fig. 37.—Types of Blading used in Richardsons, Westgarth Impulse-reaction Turbines

that the general rotor construction is not dissimilar to that commonly used in multi-stage impulse practice.

The various types of impulse and reaction blading adopted by this firm are shown in fig. 37. In their latest machines the impulse blades are of rustless iron or steel. The reaction blade material is bronze, and the blades are tipped at the ends and single- or double-laced according to length.

The design of the glands, dummies, &c., follows Messrs. Parsons' practice, from whom Messrs. Richardsons, Westgarth have held a licence since 1903.

The governing is of the oil-relay type, the control being of the combined throttle and nozzle type.

A 25,000-Kw. machine is installed in the Stuart Street station of the Manchester Corporation Electricity Department, and has been in operation for about three years. Steam consumption tests carried out after twenty-two months' running showed no fall in economy against the first tests carried out fourteen months previously.

CHAPTER X

The Ljungström Turbine

The Ljungström turbine represents a radical departure from general turbine practice, and is characterized by its many original and ingenious mechanical features. This design was invented and developed by two brothers, Messrs. F. and B. Ljungström of Stockholm. The more usual

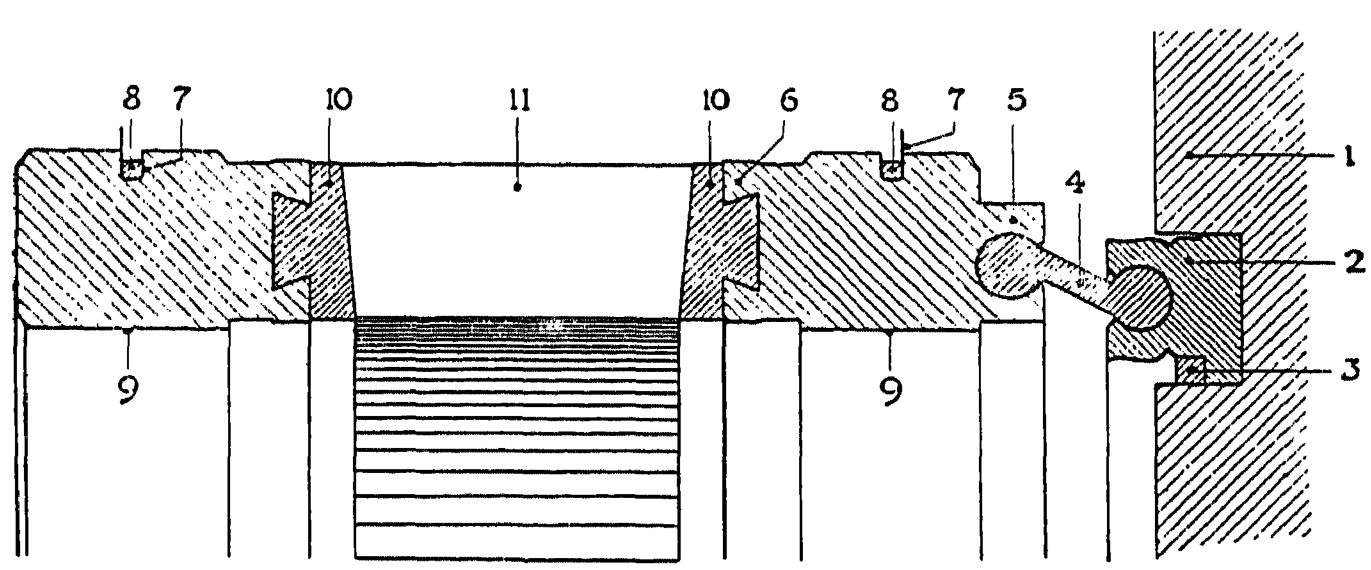


Fig. 38.—Section of Blade Ring

principle of stationary and revolving blade systems is here discarded in favour of two revolving blade systems or rotors rotating in opposite directions. This principle secures the advantage of a relative blade speed equal to twice that of the actual speed of revolution. Each rotor consists of a shaft end carrying a disc on the face of which the horizontal blade rings are mounted concentrically. The blade ring spacing is so arranged that the rings of one rotor fit between those of the other. The blades of the two rotors are set in opposite directions, so that the direction of flow of the steam is reversed in alternate stages.

The steam is admitted to the centre of the blade system and expands radially outwards; in other words, the turbine is of the radial-flow type.

The turbine casing is therefore only subject to the temperature and pressure of the exhaust steam, and does not need to be lagged. This, together with other features of the design, makes it particularly well adapted for utilizing steam at high pressures and temperatures. The Ljungström turbine has

been manufactured and developed in this country by the Brush Electrical Engineering Company, of Loughborough, since 1913.

Fig. 38 shows a section of an individual blade ring. The blade strip is milled out of solid nickel steel bars, and the individual blades are cut to required length with a taper projection at either end. These projections fit into corresponding notches formed in two welding discs. Thin sheet-iron strips are temporarily inserted between individual blades to ensure correct spacing and angles, and the taper ends are then welded into the discs. The sheet-iron strips are then removed and the discs cut down to dovetailed profile rings (10). The strengthening rings have the initial shape shown in fig. 39 (12), and the rolling edges (6) are rolled on to the dovetail. The same method of rolling is adopted for holding the strengthening rings to the dumb-bell section expansion ring (13). This ring has two functions. Firstly it gives the necessary play to the blade ring to allow for temperature expansion, and the second purpose is served by the narrow section which reduces heat conduction from the blade rings to the disc. The expansion rings in turn are held in seating rings (2) which are caulked into knurled slots cut in the discs (1), (fig. 38).

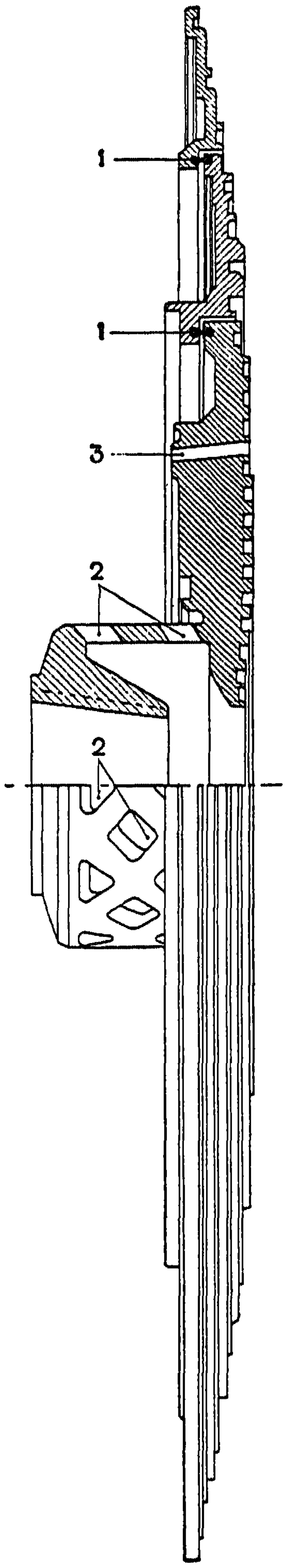


Fig. 40.—Disc and Hub for 1000-Kw. Ljungström Turbine

Steam leakage between stages is reduced to a minimum by means of the nickel clearance strips (7), which are caulked into the strengthening rings and which project within a few thousandths of an inch of the strengthening rings of the next blade ring.

Where the required blade area calls for blade lengths which are considered excessive for a single blade ring, a number of blade sections are connected in parallel, and for turbines of larger outputs of 5000 Kw. and upwards the rotors are provided with an axial-flow blade ring, the steam passing through guide blade rings before entering this stage. The radial-flow blading is designed on the reaction principle, but where a final axial-flow stage is adopted this is built on the impulse principle.

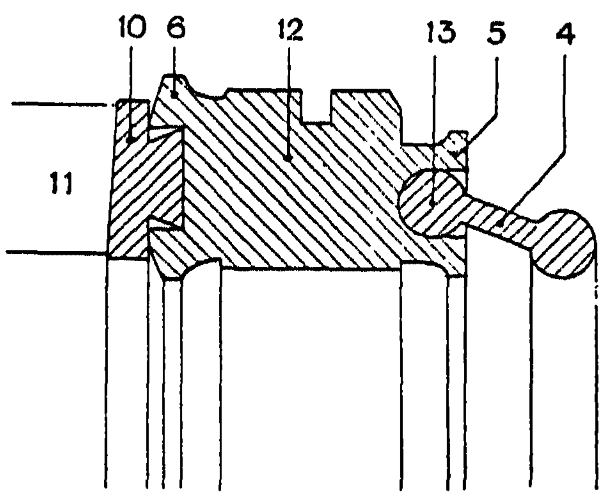


Fig. 39.—Strengthening Ring

Fig. 40 illustrates a disc of a 1000-Kw. turbine. The disc is built up in sections connected by means of dumb-bell section rings (1) so as to eliminate distortion, and one face of the disc is grooved to receive the caulking rings by means of which the blade rings are attached to the disc. The other side of the disc is arranged to receive the labyrinth disc referred to later, and has

an extension forming a hub. The hub is provided with a number of openings (2) through which the steam is admitted to the centre of the blade system. The bore of the hub is tapered, and the disc is connected to the alternator shaft through an extension shaft, to which it is secured by means of a number of round keys which are retained by a locking device screwed into the end of the shaft. From this it will be understood that each turbine rotor is overhung from the end of its alternator shaft. The extension shaft, which lies within the turbine casing and is subject to heating due to the steam, is made hollow, so that the fluctuations in the temperature of the shaft and the hub will be practically simultaneous, and relative movement will be avoided.

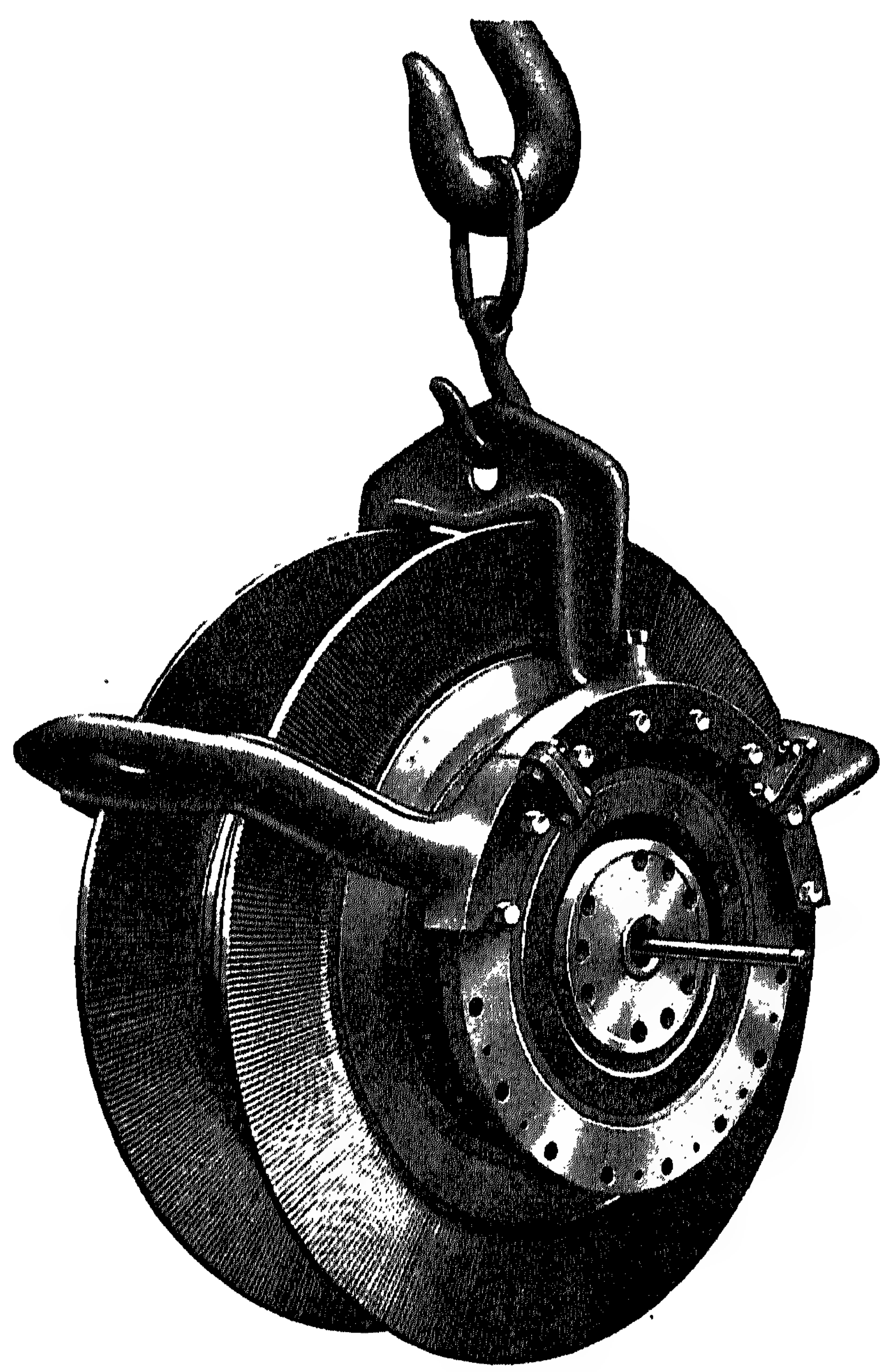


Fig. 41.—Special Lifting Stirrup for 5000-Kw. Ljungström Turbine Rotors

The passages (3) through the disc are provided to allow for the admission of steam at an intermediate stage of the blade system when the turbine is operating under overload conditions.

Fig. 41 illustrates the two rotors of a 5000-Kw. Brush Ljungström turbine, designed to run at 3000 r.p.m., and shows the special lifting gear. The internal parts of the turbine are made accessible by removing the upper portion of the turbine casing. After the set-screws have been removed, the couplings between the generator and the turbine wheels are disengaged by a slight axial movement of the alternator rotors, for which provision is made. By the aid of the lifting gear which is screwed to the steam chests these are lifted, together with the turbine wheels, thus making the details of the turbine accessible for inspection. The two rotors are locked together by means of segments fitted into dovetailed grooves in the discs so as to prevent damage to the packing strips and the blade rings. After removal from the turbine casing a centre mandril is inserted, and the turbine wheels with their steam chests are clamped in an iron frame before removing the lifting arm.

Fig. 42 shows a half section through a 5000-Kw. turbine running at 3000 r.p.m. The left-hand portion of this section represents the bottom half of the design inverted.

The steam passes through a throttle valve and enters the turbine case through a pipe leading through the exhaust chamber. This pipe is divided within the casing into two, each branch leading to a separate stationary steam

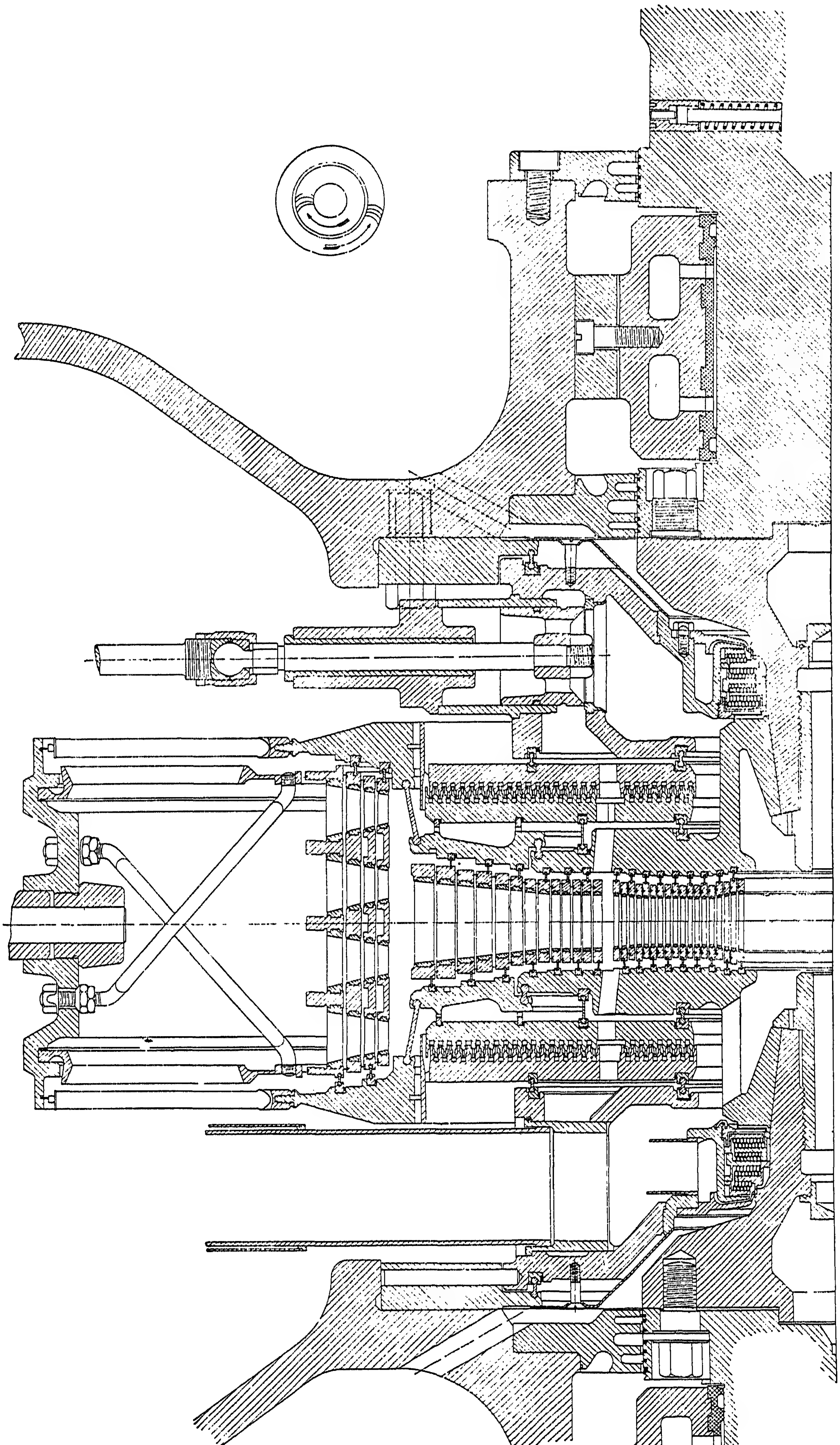


Fig. 42.—Ljungström Turbine

Half Section through 5000-Kw. Ljungström Turbine. Left of centre line, bottom half. Right of centre line, top half.

chest formed circumferentially round the turbine shaft and containing the overload valve, which is mechanically controlled and admits high-pressure steam to an intermediate stage of the turbine. The main steam supply passes through the ports in the hubs previously referred to. It will be seen that the last four radial blade elements each consist of four blade rings in parallel, and the final stages are of the axial-flow impulse type as already referred to. The guide blades are mounted in separate cages, and the illustration shows the means of mounting the cages in the turbine. The method adopted is designed to relieve the guide blades of temperature stresses.

Steam packing is required to prevent excessive leakage along the shafts to atmosphere, and between the rotating turbine discs and the stationary steam chests. The shaft leakage is checked by compact labyrinth glands formed between the steam chest and the shaft ends.

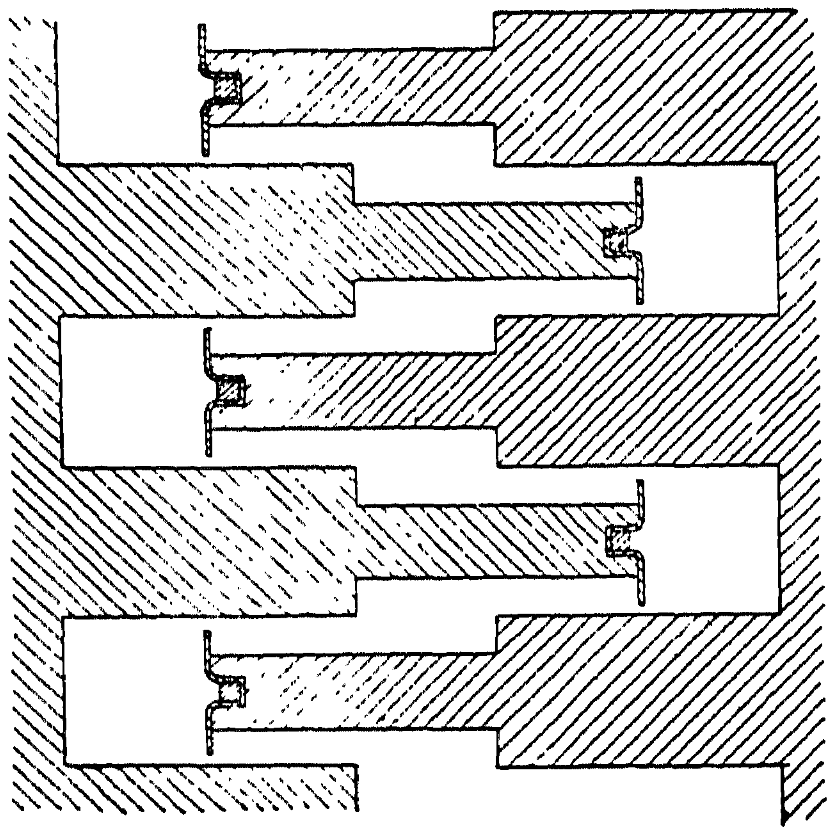


Fig. 43.—Labyrinth Packing for Ljungström Turbines

The packing to prevent leakage to the condenser consists of grooved discs fixed to each blade disc and one to each steam chest. Fig. 43 shows the arrangement of grooving in opposite discs in relation to each other, the constriction being formed by nickel strips caulked in the projecting disc face and bent over to give a fine running clearance against the corresponding opposite walls of the groove within which the disc projection runs. In earlier designs these labyrinth packings were designed to balance the thrust due to the pressure acting on the blade discs, but according to latest practice this thrust is taken up by Michell thrust bearings on the main alternator shafts, this arrangement greatly simplifying the labyrinth disc design.

The main governor is of the centrifugal type, and actuates the throttle valve through relay gearing.

The turbine casing is split along the horizontal plane, and is bolted at either end to the main alternator shells. The turbine casing forms the bearing supports for the two inboard alternator bearings, and is so shaped as to form the adjoining end covers for the alternators. The exhaust flange is bolted directly on to the condenser inlet opening, the condenser acting as the main supporting agent for the whole unit, so that condenser foundations alone are required. The light weight of the turbine, coupled with the fact that no part of the casing is subjected to high-temperature steam, makes this arrangement practicable.

CHAPTER XI

Special Types: Exhaust-steam, Mixed-pressure, Back-pressure, and Pass-out Turbines

The previous chapters have dealt mainly with steam turbines of the high-pressure condensing type. Four other types, differing in the pressure range utilized, are as follows:

1. Low-pressure or exhaust-steam turbines with steam admission at about atmospheric pressure and exhausting into a condenser.
2. Mixed-pressure turbines combining the functions of (1) with those of a high-pressure condensing turbine.
3. Back-pressure turbines with steam admission at high pressure, and exhausting against atmospheric pressure or higher pressure.
4. Pass-out turbines, which combine the functions of (3) with those of a high-pressure condensing turbine.

Exhaust-steam Turbines.—The most usual function of a low-pressure turbine is to utilize the steam exhausted from one or more non-condensing reciprocating engines. If these engines, either from their design, the nature of their work, or the low initial steam pressures, are uneconomical, as is very often the case, then the exhaust steam may yield more power in the turbine than the output of the engines. If, as in a few cases, no extra power is required, then the result of adding the exhaust-steam turbine is to reduce the demand for steam by 50 per cent or more.

Taking next the case of a reciprocator working, condensing, and having an output of 1000 b.h.p. and a steam consumption of 16 lb. per brake horse-power hour. If this engine is arranged to exhaust to atmosphere, which can usually be effected without loss of output, and if its steam consumption is thereby increased by 25 per cent, i.e. to 20 lb. per brake horse-power hour, then an exhaust-steam turbine having a consumption of, say, 25 lb. per brake horse-power hour will give an output of $\frac{1000 \times 20}{25} = 800$ h.p. The over-all steam consumption is thus reduced from 16 lb. per brake horse-power hour to 11·1 lb., an improvement of just over 30 per cent. If the additional power made available can be effectively utilized, the capital expenditure involved will be fully justified.

Sir Charles Parsons recognized the savings to be obtained in this manner, and built a number of exhaust turbines in the nineties of last century.

Exhaust-steam Accumulators.—The exhaust turbine, however, received its greatest impetus from the invention by Professor Rateau of the exhaust-steam accumulator, the interposing of which between engines and turbine makes it possible for a low-pressure turbine to draw a constant supply of steam while the reciprocating engines are exhausting widely fluctuating

steam quantities, as in the case of rolling-mill and colliery-winding engines, and due to this invention many collieries and steelworks have been enabled to obtain an electric supply at a very low coal cost.

The accumulator consists, in principle, of a vessel containing water and fitted with a steam inlet branch connecting to the exhaust-steam range of the reciprocating engines and steam outlet leading to the low-pressure turbine. When the engine is running, part of the steam, corresponding to the rate of consumption of the turbine, passes through the accumulator to the

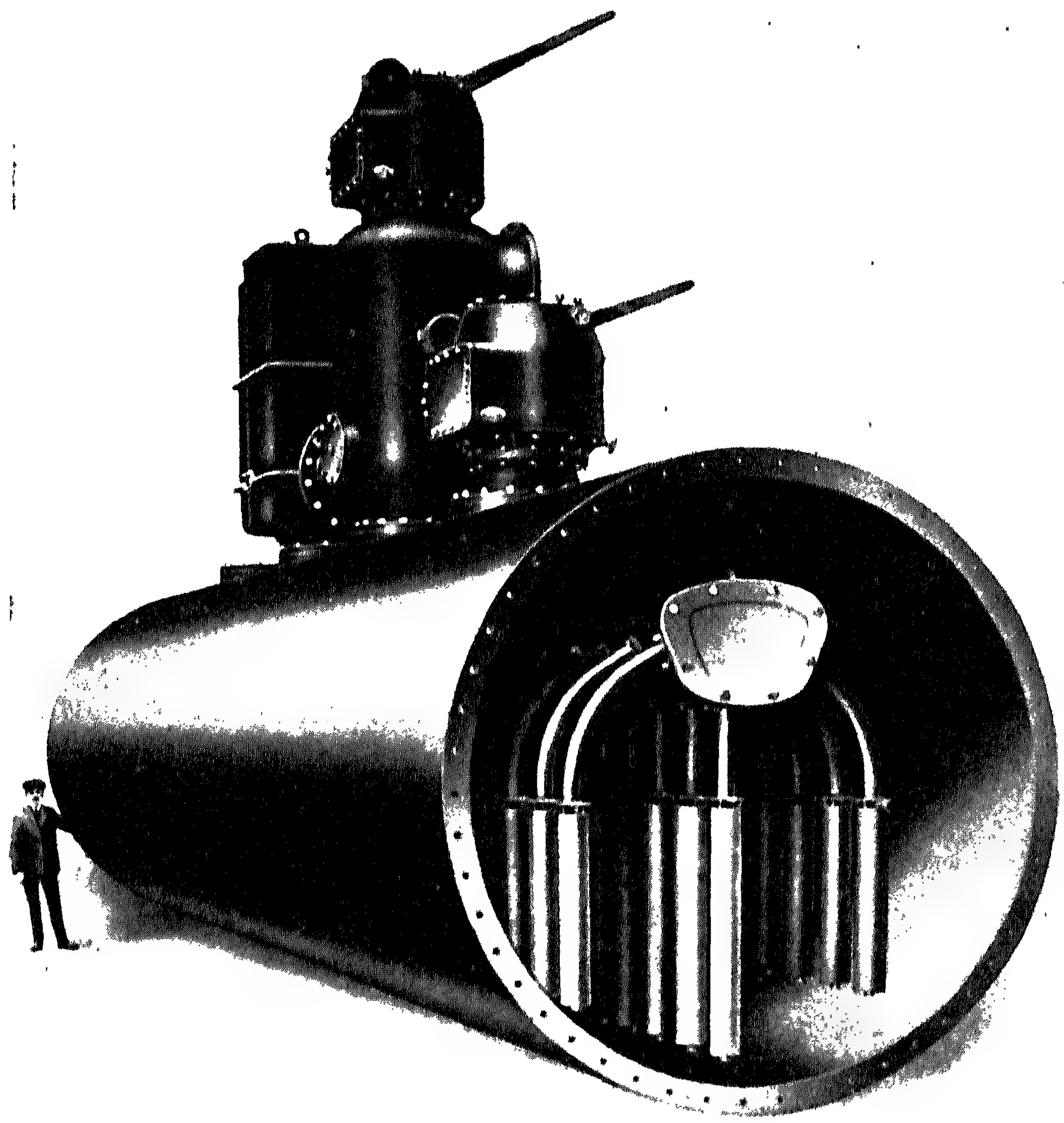


Fig. 44.—Rateau-Morison Accumulator, Cover removed showing Circulating Tubes

turbine. Any steam entering the accumulator in excess of that consumed by the turbine is condensed in the accumulator at a pressure slightly above atmospheric, the latent heat rejected by the steam being taken up by the water. When the steam supply from the engine is interrupted the pressure in the accumulator falls, due to the suction from the turbine, and the steam re-evaporates at this reduced pressure, which should be only slightly below the atmospheric.

Fig. 44 shows a Rateau-Morison accumulator, which embodies a number of improvements due to Mr. D. B. Morison. The incoming steam passes into the steam main shown in the centre of the steam space of the accumulator, and passes down the small tubes which reach to about 9 in. below the

water-level and end in a form of steam nozzle. The steam tubes are surrounded by circulating tubes which extend from the bottom of the accumulator to the water-level. The steam issuing from the nozzles causes a rapid circulation of the water, thereby ensuring that the whole bulk of the water is effectively utilized. At the same time the arrangement provides an efficient means of condensing the surplus steam. The openings at the top of the circulating tubes are all arranged in one direction, so that a current is set up which forces all oil and floating impurities towards the end of the accumulator into a collecting chamber from which the oil is drained away. This arrangement enables separate oil separators to be dispensed with.

A by-pass valve is fitted between the engine exhaust and turbine inlet pipes, so loaded as to open when a sudden rush of steam occurs, and thereby avoiding any undue rise of back pressure against the engines. The accumulator requires to be effectively lagged to reduce radiation losses to a minimum. The size of an accumulator of a given type and to deal with a given mean steam quantity depends on the rate of fluctuation of the steam supply and the length of time during which regeneration has to take place, i.e. the period during which the steam supply rises or falls below the rate of demand from the turbine.

Mixed-pressure Turbines.—An accumulator designed to bridge over periods of stoppage of several minutes becomes excessive in size and cost, and if, as frequently happens, the load on the turbine is maintained over prolonged periods, during which no exhaust steam supply is available, it is necessary for a live steam supply to be admitted to the turbine through a reducing valve. This method of working is inefficient, and led to the introduction of mixed-pressure turbines, which have largely taken the place of exhaust turbines. A mixed-pressure turbine consists of an exhaust turbine and an auxiliary high-pressure turbine within one casing, and in practice it takes the shape of a standard high-pressure turbine with an enlarged low-pressure end.

In comparison with an exhaust-steam turbine, a mixed-pressure turbine is naturally somewhat lower in efficiency under equal conditions for equal blading, below atmospheric pressure stage when working on low-pressure steam, as the high-pressure stages are rotating without doing useful work, and consume power in the shape of windage losses. The capital cost is naturally also somewhat higher. The advantage of running more efficiently when no exhaust steam is available or when the supply of exhaust steam is insufficient to meet the load on the turbine usually more than compensates for these disadvantages.

Properly to perform the regulation of the steam supply to a mixed-pressure turbine calls for a valve gear to meet the following requirements.

First, it should allow the turbine to utilize all available low-pressure steam before admitting high-pressure steam.

Second, its operation should not interfere with the running speed of the turbine and vice versa.

These conditions are fully met by designs based on Professor Rateau's

patent system of governing mixed-pressure turbines. Fig. 45 shows isometrically the design adopted by the Metropolitan-Vickers Company based on this patent. The valves are operated by oil pressure, as in the case of their high-pressure turbines.

The low-pressure admission valve is suspended from one end, and the high-pressure admission valve from the other end of a cross lever N, pivoted at one end of a piston rod working in the cylinder C. The cylinder D contains a spring pressing upon the piston on the valve rod, tending always

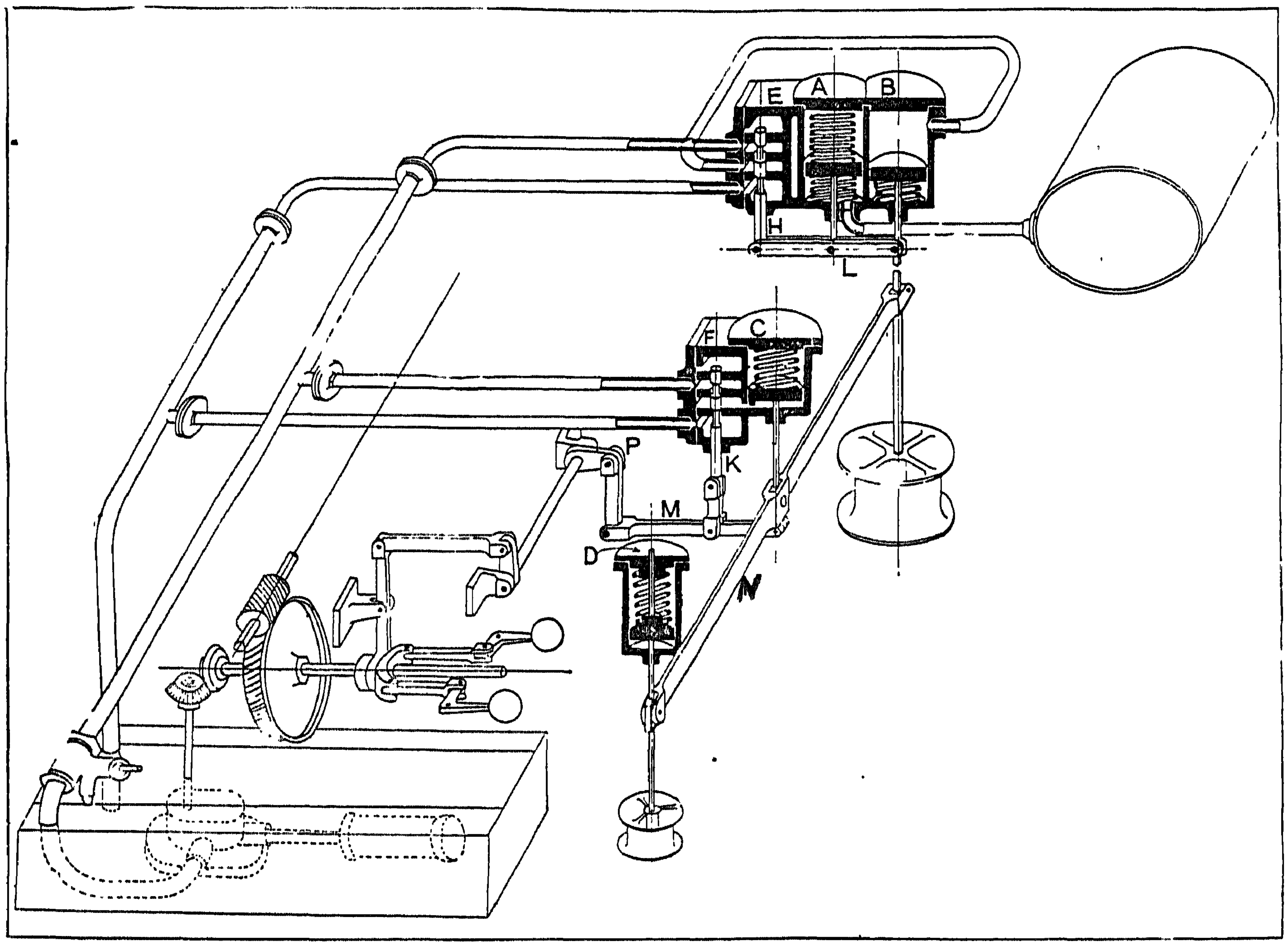


Fig. 45.—Isometric View of Valve Gear of Metropolitan-Vickers Mixed-pressure Turbine

to keep the high-pressure admission valve on its seat. It will be observed that the plunger rod in the cylinder B is cut off immediately below the lever L, and is not continuous with the suspension rod of the low-pressure admission valve. The low-pressure steam main is connected by a small pipe to the underside of the piston in the cylinder A, springs above and below this piston being selected for the particular limits in the low-pressure main through which the governor gear is required to operate. The oil is continuously pumped at about 50 lb. pressure to the chests E and F, any surplus returning to the oil tank through the relief valve shown. Assuming that a drop of pressure takes place in the low-pressure steam main, the equilibrium in the cylinder A is upset and the lever L is rocked downwards with its fulcrum on the centre line of the cylinder B. This pulls down the valve H and opens up the oil pressure through the port shown to the top of the piston in B. The effect of this is to cause the piston to descend against the spring beneath

it and rock the lever L with the centre line of A as a fulcrum. The valve H is therefore returned to its original position, cutting off further oil pressure from the top of the piston in B. The amount which the latter moves is therefore dependent upon the movement of the piston in A, which in turn is dependent upon the fluctuation in pressure in the low-pressure steam main.

The piston in B moving downwards presses upon the top of the valve rod suspending the low-pressure re-admission valve, pressing it either on to its seat or advancing it towards its seat to such an extent as to throttle the steam supplied through the low-pressure valve and maintain the pressure in the low-pressure steam main. For any load the closing (either partial or absolute) of the low-pressure admission valve is simultaneously followed by an opening of the high-pressure admission valve, and since these valves are proportioned to pass amounts proportional to their relative steam values, the speed is maintained constant. As soon as the pressure rises in the low-pressure steam main the reverse operation takes place, the low-pressure admission valve being opened and the high-pressure closed. A change of load will, of course, affect the speed of this turbine just as in any other turbine or steam engine, the governor balls moving inwards with an increase in load due to a slight decrease in speed. It will be seen, by following the movements of the levers shown, that the connecting rod P will be depressed with an increase in load, rocking the lever M, with the centre line of C as a fulcrum, thus pulling down the valve K, and opening up the oil supply under the piston in C. As this piston rises against the spring above it, it opens either the high- or low-pressure admission valves, one or both of which may be in operation, and at the same time rocks the lever M again with the centre line of the connecting link P as a fulcrum, raising the valve K and shutting off the oil supply. It will be seen from this that the position of the piston in C, and hence the position of the valves, is dependent upon the load on the turbine.

Back-pressure Turbine.—A back-pressure turbine can be considered as the converse of an exhaust-steam turbine in construction as well as in its application. Thus, as an exhaust-steam turbine is, in regards to casing and blading, the low-pressure portion of a standard high-pressure turbine, so a back-pressure turbine takes the general form of the high-pressure half of the same machine, although usually in a simplified form.

Its use arises where a demand for electrical or mechanical power coincides with a demand for low-pressure steam, which may be needed for most varying purposes, such as factory heating, boiling and drying in paper-mills, chemical-works, and the like, or for feed heating.

The difference in coal cost between raising steam at low and high pressures and temperatures being a fraction of the total, it is economical to raise steam at a comparatively high pressure and temperature, and to allow an engine or turbine to convert the available difference into mechanical energy before the steam is passed into the heating system.

The actual coal cost chargeable to the power generated by the engine or turbine, assuming equal boiler efficiency when raising steam under either

condition, is represented by the difference in the heat contents at the steam-engine stop valve and exhaust divided by the boiler efficiency expressed as a percentage.

The thermodynamic efficiency of the machine under these conditions of working cannot be measured by the ratio of work performed expressed in heat units to heat units available in adiabatic expansion, but as the ratio of work performed to the sum of work performed and heat lost in direct radiation and the heating of lubricating oil. For all losses resulting in re-heating the steam can be utilized in the heating system. It follows that under such conditions the measurement of the steam consumption of the turbine or engine in pounds of steam per horse-power hour does not give an indication of the efficiency, and although a back-pressure reciprocating engine will generally show to advantage in the matter of such steam consumption, the turbine may be the more economical prime mover on account of its reduced radiation losses. The absence of oil in the exhaust steam is often a deciding factor in favour of a turbine.

Fig. 46 shows a single-stage impulse turbine of Messrs. Fraser & Chalmers' manufacture, specially designed for small outputs and well adapted for use as a back-pressure turbine for driving pumps and other auxiliary machinery where it is desired to use the low-pressure steam for feed-heating purposes. It will be seen that the machine illustrated is compounded for velocity, although under certain steam conditions a single blade row is employed.

The casing consists of two high-grade cast-iron portions jointed horizontally and bolted together. The bottom half of the casing is provided with a substantial foot on either side, and also carries the bearing brackets which are bolted to the casing. The steam admission and exhaust openings are arranged in the top half of the casing.

The type of blades, design of bearings, and other essential features follow the makers' standard practice already described, a special feature of the design being the method of governing.

Close to the main oil-pump at the high-pressure end of the shaft a similar pump is fixed which performs the functions of a governor. The supply from this pump is taken to a governing cylinder containing a piston loaded by an adjustable spring. As the oil supply of the governor pump is practically nil, the delivery pressure varies with the square of the speed of the turbine. This variation in oil pressure is utilized to operate the piston in the governing cylinder, which in its turn moves the piston valve admitting oil to or releasing oil from underneath the power piston of the governing valve. The speed of the turbine can be varied by altering the tension of the spring in the governing cylinder by means of a hand-wheel. A lever connects the spindles of the governing piston, piston valve, and power piston, and therefore to every position of the governor piston corresponds a definite lift of the governing valve.

The retention of relay governing is somewhat exceptional in a small-power turbine of this kind and ensures very close governing.

Pass-out Turbine.—In many instances the demands for power and low-pressure steam for heating purposes do not coincide sufficiently closely in regard to time or magnitude to obtain satisfactory working results with the use of a simple back-pressure machine, and it is then necessary to

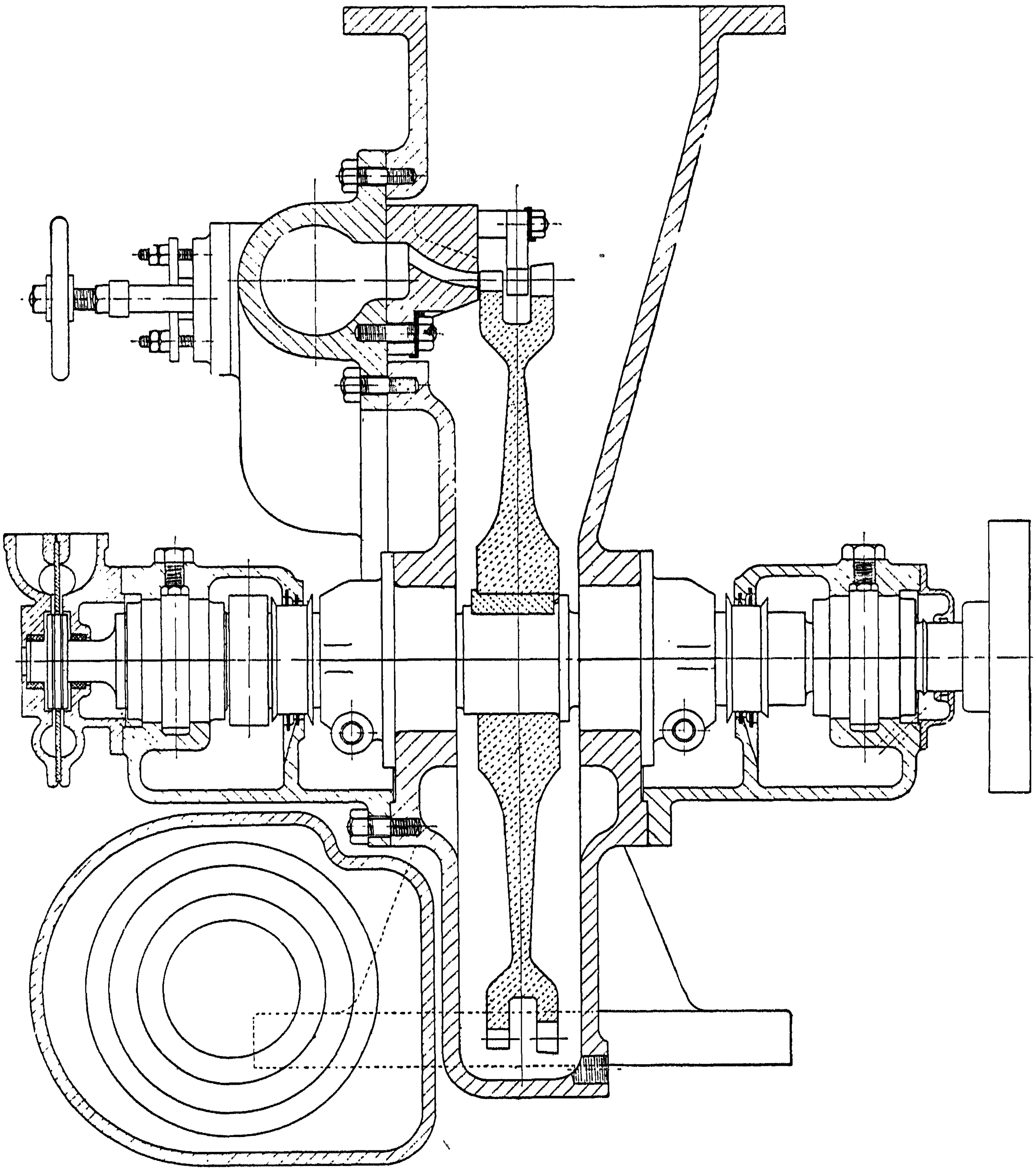


Fig. 46.—Section through Fraser & Chalmers' Single-stage Back-pressure Steam Turbine

adopt a pass-out turbine, which, in principle, represents the converse of a mixed-pressure turbine and consists of a back-pressure turbine with auxiliary low-pressure stages. The installation must then include a condensing plant. Such a turbine should be capable of supplying any quantity of low-pressure steam required by the heating system and consistent with its loading. Where the heating load is correspondingly less than the power load, then the additional steam admitted to the turbine must pass into the

low-pressure stages of the turbine and the condenser without interfering with the low-pressure steam supply. If the demand for low-pressure steam is interrupted, the turbine should work as a simple high-pressure condensing machine. Suitable valve gear is required to ensure that the turbine adapts itself to the varying conditions under which it may be required to work.

Fig. 47 shows a section through a Curtis pass-out turbine built by Messrs. The British Thomson-Houston Company. It will be seen that the general construction is similar to that of their high-pressure condensing turbines, with the addition of the necessary valve gear to control the admission of steam

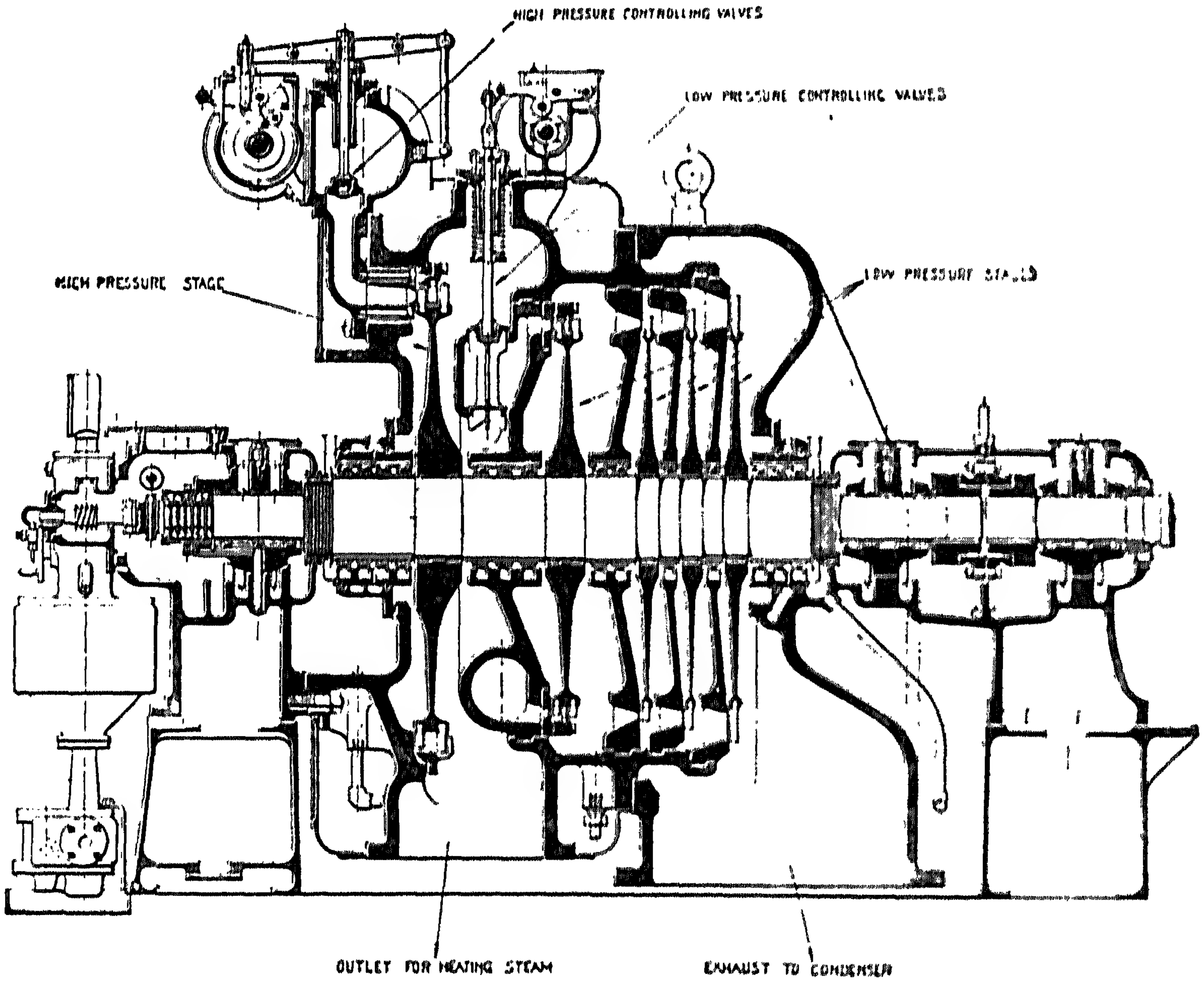


Fig. 47.—Sectional Elevation of Curtis Pass-out Turbine, built by The British Thomson-Houston Co., Ltd.

to the low-pressure stages, and of a suitable outlet provided for the supply of heating steam.

The British Thomson-Houston Co., Ltd., have developed a patented automatic valve gear which enables a constant speed to be maintained with a constant load independently of any fluctuation in the demand for heating steam.

This gear is generally very similar to the valve gear used on mixed-pressure turbines, the low-pressure valves being multiple equilibrium valves operated by a rotating servo-motor and cam gear, with a pressure regulator of similar construction to that used on low-pressure steam.

The high- and low-pressure valves are connected to the speed governor, to the pressure regulator, and to each other in such a way that change of speed produced by a change of load will move both high- and low-pressure valves in the same direction, either to open or to shut as may be required. On the other hand, an alteration in the quantity of heating steam will produce

a slight variation of pressure, which by means of the pressure regulator will cause the high- and low-pressure valves to move in opposite directions, i.e. low pressure to shut and high pressure to open or vice versa, thus maintaining a practically constant pressure without the speed governor being affected.

CHAPTER XII

Marine Turbines

It was recognized very early in the development of the steam turbine that it would be a suitable prime mover for the driving of the propellers of steamships. The principal difficulties to be overcome were connected with the fact that propellers and turbines have their highest efficiency at widely different rates of revolution.

The adoption of the turbine drive for ship propulsion was a revolutionary departure from previous marine practice.

Sir Charles Parsons exercised to the full his genius for investigation and invention in endeavouring to solve some of the problems involved, but it was some long time before his experiments eventually came to fruition in the *Turbinia*, the first turbine-engined ship. This was of torpedo-boat type, and had three turbines, each on a separate shaft, and capable of developing a total of 2000 equivalent indicated horse-power.

Before satisfactory arrangements were evolved numerous different designs of propellers had to be tried. At first several were placed on each shaft, but experience narrowed the solution down until it was found that single propellers on each shaft were preferable, and such is the present-day practice.

Since the evolution of the *Turbinia* the application of the steam-turbine has made such rapid growth that at the present day more than 16½ million horse-power of direct-coupled units and about 18½ million horse-power of geared units have been made in this country alone and are in use in marine work.

After the *Turbinia*, two turbine-driven destroyers, the *Viper* and the *Cobra*, were commenced in 1898.

Both the *Viper* and the *Cobra* achieved remarkable results, but unfortunately both were lost, though the loss was in neither case due to the turbine installation.

Although it was recognized that the turbine drive was successful under the conditions of naval service, the mercantile marine shipowners were not immediately convinced that it would be so if applied in the merchant service. In the navy efficiency and reliability are the first considerations, whilst cost of installation and running expense are subordinate.

In the earlier vessels the efficiency of the turbine when running at low loads was not good, and shipowners were not inclined to take the results

attained by the *Turbinia*, the *Cobra*, and the *Viper* as indicative of success for the conditions obtaining in the merchant service.

In mercantile steamships high speeds are not generally desired, excepting for fast passenger boats, and, as has been said, the turbine gave its best efficiency at high speeds. On the other hand, a merchant vessel required a practically constant speed rate, and therefore turbine machinery could be designed to give highest efficiency for a fixed speed. It was found, however, that even with improved propeller design the lowest speed that was satisfactory for a direct-coupled unit was about 18 knots.

The main idea of the shipowner is ultimately one of economy, and it was only after some years that the turbine was eventually adopted for the merchant service.

The first applications were for high-speed vessels for passenger traffic, and from that intermediate speed vessels were fitted. The application in these was entirely successful, but it was not until the introduction of of mechanical gearing that the problem of fitting turbine drives for low-speed vessels can be considered to have been successfully dealt with.

The first turbine merchant steamer was the *King Edward*, which was a joint enterprise of Captain John Williamson, Messrs. Denny, and the Parsons Company. This boat was built in 1901. One high-pressure machine with four expansions was fitted to the centre shaft. The steam from this went to two low-pressure machines, one at each side, and in the same casing as these were incorporated the astern turbines. Originally two propellers were fitted to each of the wing shafts, but later a single propeller was fitted to each shaft. The actual turbine construction differed somewhat from that manufactured for war vessels. Heavier construction was used, and the turbine shaft did not go right through the drum, but was shrunk into wheels carrying the drums. This boat was put in service in the estuary of the Clyde, doing about 180 miles steaming per day.

It was found that this vessel not only had a greater mileage and a higher speed, but that it burnt less fuel than other vessels on the same service, and the coal consumption taken over a period of years was found to be nearly 20 per cent less than that of vessels of similar size and speed but fitted with compound engines driving paddle-wheels.

The success of this turbine-driven boat was followed by the adoption of the drive for a number of vessels engaged in the cross-Channel service.

One of the great objections in the early evolution of the turbine for ship-drive was its non-reversibility, and in some of the earlier installations it is certain that the astern turbines were not powerful enough for their work. Nowadays, as a general rule, they are made much larger, and are usually capable of developing at least 50 per cent of the maximum power of the ahead turbines. The turbine has a greater power of stopping or reversing the direction of the ship than the reciprocating engine. Since at the moment of reversal the ahead turbine is still revolving the astern turbine in the ahead direction, the effective torque exercised by the steam admitted to the astern turbine is considerably greater than under normal running conditions, and

thus the breaking or retardation effect is greater. In turbine steamers the direction of rotation of the shaft has been changed within 12 sec. of the giving of the signal.

Late in 1904 the Cunard Company resolved to adopt the turbine drive in their latest Atlantic liners. This decision, coupled with the important recommendation of a special committee appointed by the Admiralty to adopt the system for all armoured ships, was the means of finally crystallizing opinion in favour of the system. The *Carmania*, of the Cunard Company, was built with turbine machinery in 1904, and, at the same time, a sister ship, the *Coronia*, was built, but fitted with reciprocating engines.

The turbine installation proved entirely satisfactory, and showed great improvement in comparison with the sister ship, and in the ensuing year many further applications were made, not only for high-speed steamers, but for vessels for other services.

Two of the largest ships built up to 1922 driven by turbines were the *Mauritania* and *Lusitania.*

In connection with these installations much research work was done, both as regards the turbines themselves and in connection with the propellers. Four shafts were adopted, each to transmit up to 20,000 h.p., and each with one propeller. Six turbines were installed, two high-pressure and two low-pressure ahead and two high-pressure astern. The two outer shafts were driven by the high-pressure ahead turbines and the two inner shafts by the low-pressure ahead turbines. The high-pressure astern turbines were placed forward of the two low-pressure on the centre shafts. The two condensers were each in a separate compartment abaft the low-pressure ahead turbines, and the auxiliaries were situated still farther aft. The rotor drums for the turbines for these ships were forged. This was a new departure, but it has now become general for large installations.

Geared Turbines.—In reviewing the progress made since the introduction of the turbine as a prime mover for steamships, it is remarkable that development has followed closely the lines which were mapped out in the early installations. The most radical and far-reaching innovation is the adoption of gearing.

In warship work variability of speed is required with economy at all speeds. On the other hand, the merchant steamer is always running at about its maximum speed, but that speed is not required to be a particularly high one in the average vessel.

Both the conditions in the war vessel and in the merchant ship are more easily met by the introduction of mechanical gearing, and, moreover, it is possible to run the turbine at a more economical speed, and at the same time get a greater efficiency from the propeller. In the merchant vessels one solution of the problems involved has been given by introducing combination machinery—that is, reciprocating engines for dealing with the high-pressure steam, and turbines which take the exhaust from these and expand the steam down to a low vacuum, since the turbine can make use of a very

much higher vacuum than a reciprocating engine, with a greatly increased efficiency.

Thus the *Olympic*, one of the White Star liners of the largest size—46,440 tons displacement—was fitted with four-cylinder triple-expansion reciprocating engines capable of 30,000 indicated horse-power at 75 revolutions. These exhausted into turbines developing 17,000 shaft horse-power at 165 revolutions.

The reciprocating engines exhaust to the turbines at about 9 lb. absolute pressure, and the steam is expanded down in the turbines to about 1 lb. absolute, the condensing plant being designed for a vacuum of $28\frac{1}{2}$ in. Hg. with a 30 in. barometer.

Though such installations are highly economical, it is probable that gearing will become standard practice for cargo vessels and the like on account of its greater simplicity and the smaller space occupied.

In 1909 gearing was first tried in marine work by the Parsons Marine Steam Company. In order to test the suitability of gearing for interposition between the prime mover and the propeller in ship propulsion, an old steamer, the *Vespasian*, was purchased. This ship was originally equipped with reciprocating engines, and these were replaced by a turbine installation driving through a single-reduction gearing. The gearing in the engine-room of the *Vespasian* with the cover removed is shown in fig. 48.

Particulars of this experiment were fully given by Sir Charles Parsons in a paper read by him before the Institution of Naval Architects.

The trials were entirely satisfactory, and the vessel afterwards ran regularly between the River Tyne and the Continent, carrying coal. During a period in which this vessel steamed about 20,000 miles the machinery gave not the slightest trouble.

In ships in which it is required to keep down staff and where facilities for repair are small, complications are to be avoided, and the helical spur gearing meets this consideration.

In the war vessel all the original installations were equipped with cruising turbines in addition to the installation for full-speed steaming. These cruising turbines were designed to give lower speeds, and were used in ordinary running when the high speeds were not necessary or desirable.

It was found from experience, however, that the use of cruising turbines for the low speeds was not an altogether satisfactory solution of the problem of variable speed. The extra complication was found to be a disadvantage, and in recent installations they have not been fitted. Gearing has been adopted in many cases, and exhaustive trials have shown a considerable increase in efficiency both at full and cruising speeds over direct-driven turbine vessels. The propeller efficiency, it was found, could be increased from 8 to 15 per cent, and the steam consumption bettered by about 10 per cent at full power, and by as much as 30 per cent at low powers.

Gearing was universally adopted during the war for destroyers, cruisers, battleships, and battle cruisers of the highest powers, and practically no direct-coupled units have been installed recently.

In latest designs for marine work further development has taken place in gearing. Double reduction gearing has been successfully adopted in preference to the original single gearing. Fig. 49 shows the arrangements of double-reduction gearing aboard ship.

For the mercantile vessel future development appears to be along the line of the adoption of double-reduction gearing. This is one of the most striking features in the progress of marine engineering. It permits of a greater ratio between turbine speed and propeller speed without excessive

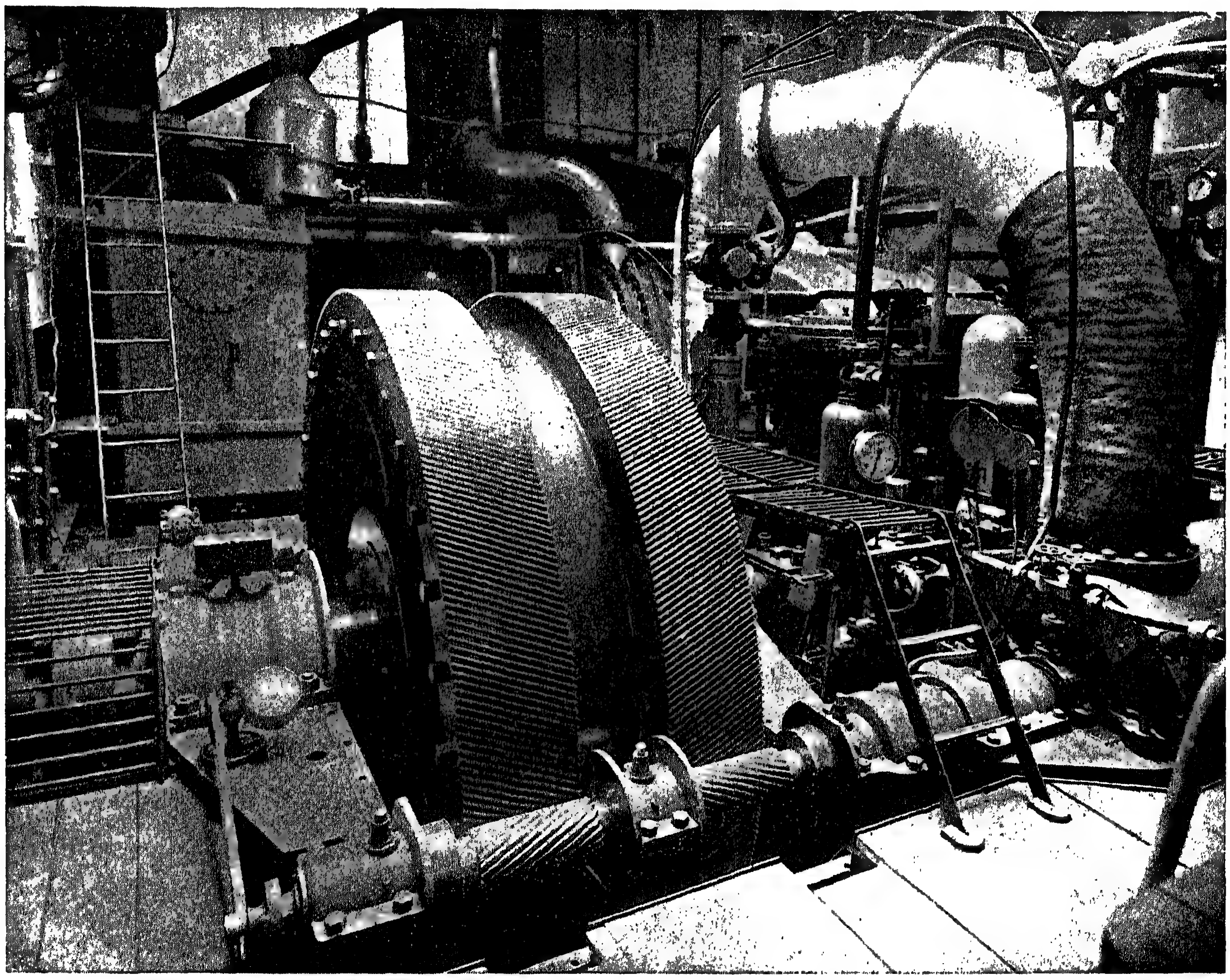


Fig. 48.—S.S. *Vespasian* Engine-room, showing Gearing with Cover Removed

gear-wheel diameter, and even in slow-speed vessels turbine and propeller can each run at their most efficient speed. It has enabled marine turbine designers to follow land practice more closely than was hitherto the case.

The development of impulse marine turbines is of later date than that of the reaction type, but has made considerable progress during the last few years, especially in conjunction with gear drives.

As an example of a typical high-speed impulse turbine adapted for ship propulsion, a brief description of that made by Messrs. The Metropolitan-Vickers Company is given.

During the war period many of the so-called standard ships were fitted with these turbines.

The high- and low-pressure turbines are arranged side by side, each driving a high-speed pinion.

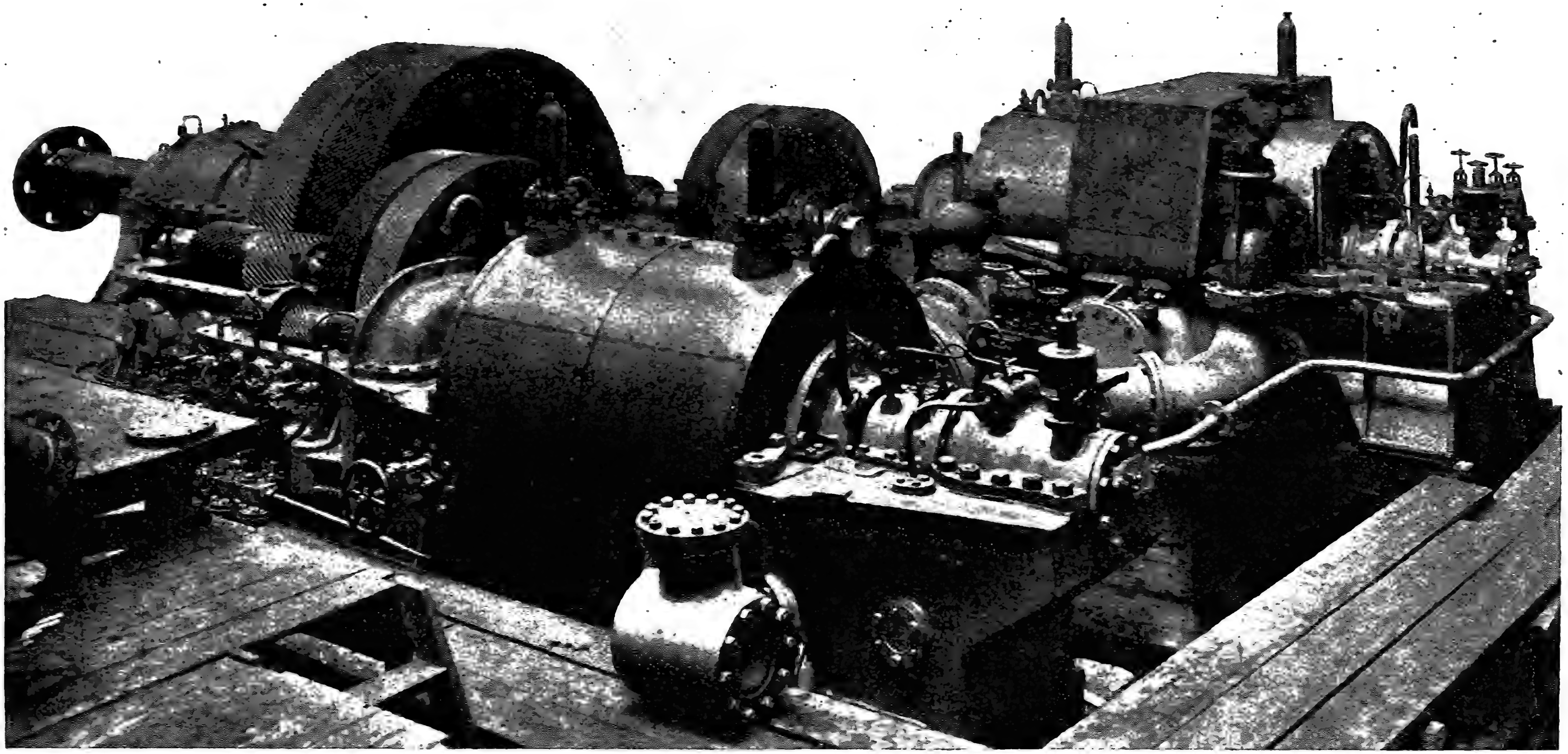


Fig. 49.—Arrangement of Turbines and Double-reduction Gear

The high- and low-pressure cylinders are each subdivided, one portion for ahead and the other for astern steaming.

On reversal the average temperature change in the high-pressure cylinder is only about 50° F. The first stage of the turbine is a two-row velocity

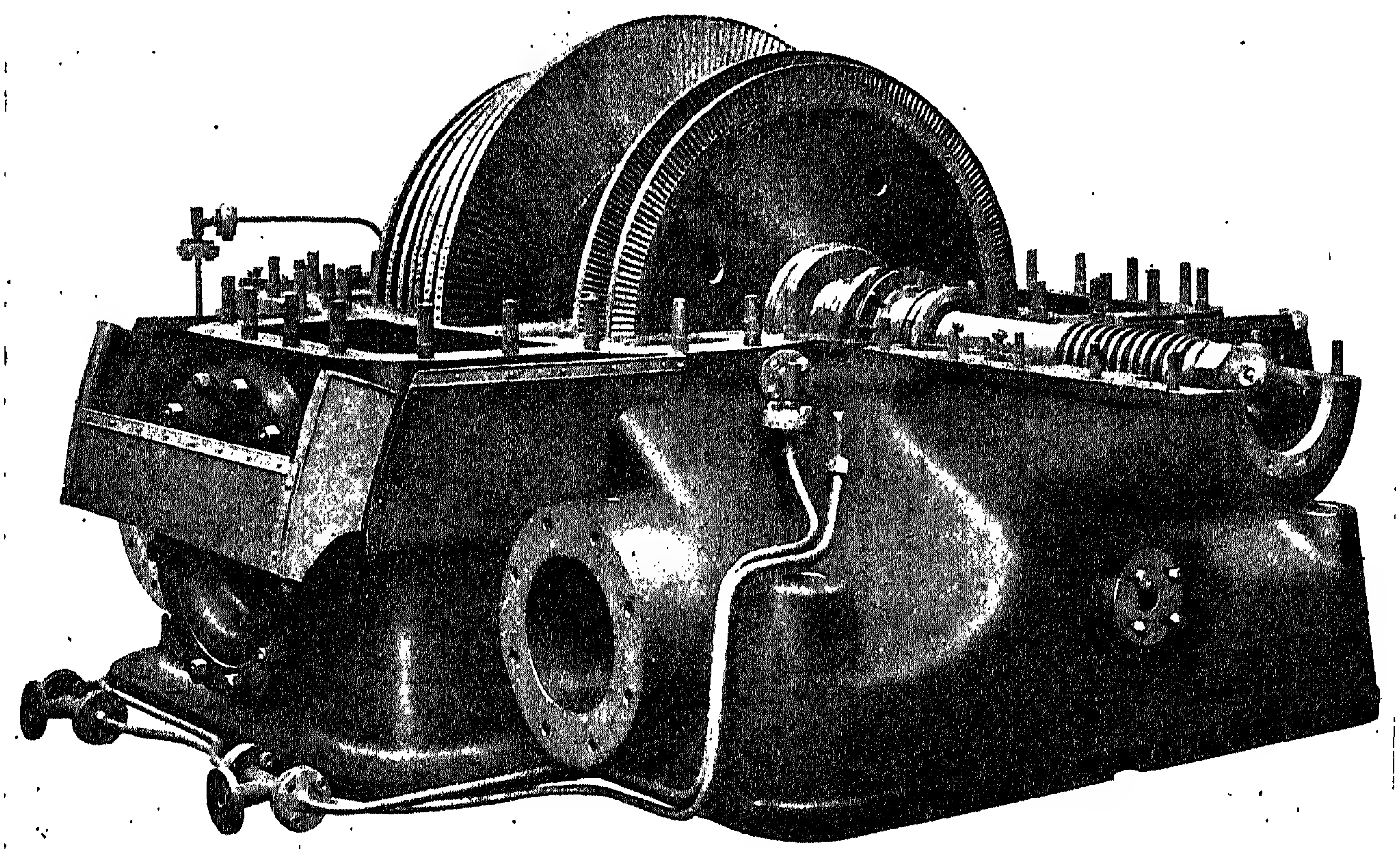


Fig. 50.—L.P. Metropolitan-Vickers Impulse Marine Turbine with Cover Removed

wheel, the remaining stages single-row wheels. The nozzles are divided into groups, so that by admitting steam to a group or combination of groups throttling losses are avoided at seven different ahead speeds.

The astern nozzles are a single group giving about 65 per cent of the normal ahead power for astern steaming. A small thrust block is provided at the governor end of the turbine, maintaining the correct register of the spindle relative to the cylinder.

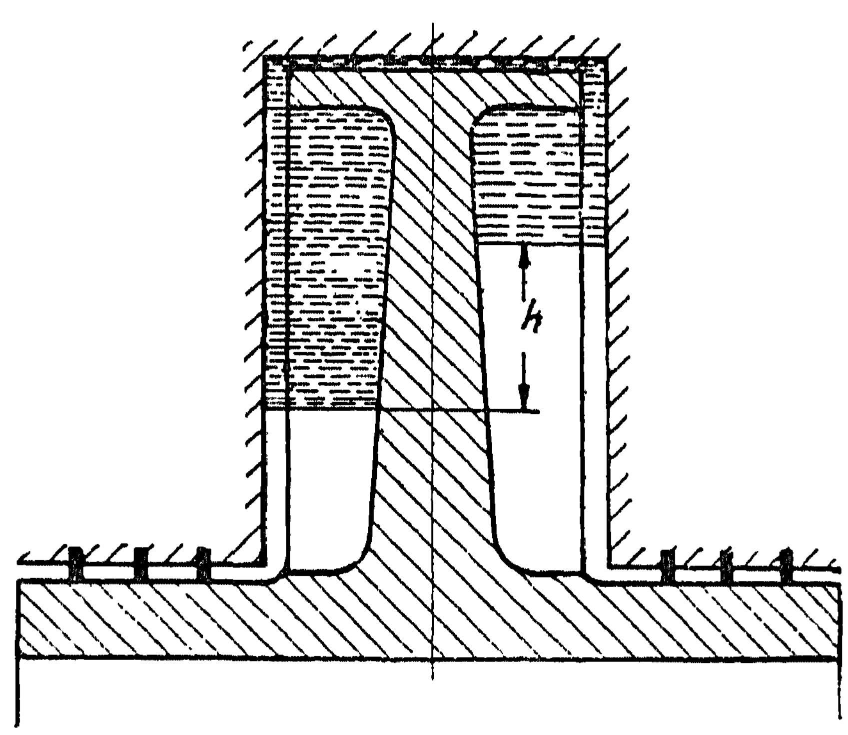


Fig. 51.—Water Sealed Gland

The high-pressure cylinder is not subjected to greater pressures than 30 to 40 lb. per square inch gauge, and these are limited to the first stage.

The cylinders are made of high-grade cast iron, and are bolted direct to the gear case, the forward ends being provided with special sliding seatings to allow for expansion.

Fig. 50 shows the low-pressure turbine with top removed.

The blading and diaphragms follow the makers' land turbine practice.

The critical speed is always kept well above the running speed.

The glands (fig. 51) are of the combined water and steam seal type.

The water seal is generally used, the steam being for manœuvring, starting, and stopping. The glands can be opened for inspection without disturbing any other part of the cylinder. Automatic overspeed devices to prevent the turbine running away are fitted to high- and low-pressure units. Relief valves are provided on high- and low-pressure cylinders to relieve excessive steam pressure in case of the breakdown of condenser auxiliaries.

Special straddle gauges are provided by means of which wear on the bearings can be readily ascertained at any time.

The turbine and gears are lubricated by a combined pressure and gravity system.

Turbo-electric Schemes.—Whilst the development of large-power gearing has gone a long way towards solving one of the main inherent problems of the application of the steam turbine to ship propulsion, the use of separate reversing turbines must still be considered a drawback. The extra capital cost which they entail is a matter of comparatively small moment. The fact that the astern turbine has to run idle when not in use entails a permanent windage loss with appreciable effect on the ship's coal consumption. Ordinary working conditions involve the necessity of the astern turbine being put on maximum load at short notice, and this condition implies a severe sudden change of blade temperature. The range of temperature which blading material can be safely subjected to under such conditions is limited, and these considerations have impeded the use of highly superheated steam on board ship, and have deprived marine turbine makers of the opportunity of availing themselves of most favourable steam economy, such as is available for land work.

If the mechanical gearing is replaced by electric reduction gearing, that is to say, if the turbine drives a generator supplying current to motors which usually drive the propellors through mechanical reduction gearing, then the reversing operation can be carried out easily on the motors without affecting the direction of rotation of the turbine.

This question of turbine propulsion of steamships with the interposition of electric generators and motors as gearing was seriously considered some thirteen years ago by the General Electric Company in America, and they found in the United States Navy Bureau of Steam Engineering an organization progressive enough to take up and try the system.

The first ship engined on this plan was the collier *Jupiter*. Up to 1919 this boat had been in commission over five years, and had been in service practically continuously without electrical trouble, thus demonstrating the reliability of the system.

The 1916 United States Navy Bill authorized four battle-cruisers and several battleships, all with turbo-electric drive. The battle-cruisers each had turbines of 180,000 h.p., and were to steam at 35 knots. The battleships required 33,000 h.p. each.

In 1919 the *New Mexico*, United States capital ship, one of the battleships, successfully completed her trials.

DATA OBTAINED FROM SHIPS ENGINED WITH IMPULSE TYPE TURBINES (METROPOLITAN-VICKERS TURBINE)

DESIGNED CONDITIONS. SERVICE FIGURES.

Ship.	Pressure, Lb. Square Inch Gauge.	Vacuum Hg. Inches. Bar, 30 in.	Superheat. Degrees F.	S.H.P.	Knots, Mean.	Mean Displacement. Tons.	Mean S.H.P.	Coal or Oil Fuel.	Tons Fuel Each Day, All Purposes.	Pounds Fuel S.H.P./hr., All Purposes.
Assiout	180	28½	100	2500	11·6	8,180	2240	Coal	34·15	1·42
Amarna	180	28½	100	2500	10·36	9,360	—	Coal	32·5	—
Mangalore . .	185	28½	Sat.	5000	12·09	16,350	3180	Oil	41·6	1·22
Manipur . .	185	28½	Sat.	5000	10·36	19,200	3010	Oil	38·6	1·195
Mathura . .	185	28½	Sat.	5000	11·19	16,600	2800	Oil	33·625	1·12
Amstelland . .	185	28	150	4500	10·93	—	2900	Oil	33·0	1·06

The propelling machinery of this ship is here briefly described.

The main features are two direct-coupled turbo-alternators and four induction motors. Auxiliary gear is comprised in two 300-Kw. direct-coupled

exciters and two motor-generator boosters. The total weight of the electrical machinery is about 500 tons.

The generators have a capacity of 12,500 Kw. each, and a voltage range by means of switching of 3000 to 4242 volts. The first voltage is used for operating all four motors from one generator, the second for operating two moto s from one generator.

The alternators are, except in the voltage variation respect, similar to land types. The motors are designed to give approximately 29,000 h.p. at 167 revolutions, and are of special design, having rotors of double squirrel cage construction.

The motor stators are provided with a pole changing winding, and are arranged for 24 and 36 poles. A complete set of electric interlocks on the gear is provided.

A number of vessels have been equipped with turbine electric drive, having Ljungström turbines as prime movers.

The first vessel of the type was S.S. *Mjölner*, equipped by the Ljungström Company of Stockholm. The principal dimensions of the ship are as follows:

Length, p.p.	225 ft.
Breadth, extreme	36 ft.
Depth, moulded	15 ft. 6 in.
Draught, loaded	14 ft. 9 in.
Displacement	2270 tons.
Block coefficient at 14 ft. 9 in. draught	0·665 in.
Gross tonnage	976.
Net tonnage	376.

The power is given as 900 b.h.p. The generating equipment consists of two 400-Kw. Ljungström turbo-alternators running at 7200 r.p.m., and exhausting into contraflow condensers capable of maintaining a vacuum of $97\frac{1}{2}$ per cent. The two 3-phase induction motors running at 900 r.p.m. are geared down to a common propeller shaft running at 88 r.p.m.

On trial the ship maintained a speed of 11·8 knots, her coal consumption being 0·89 lb. per i.h.p. hour, and 1·036 lb. per s.h.p. hour, the calorific value of the coal being given as 13,485 B.Th.U. per pound.

Three months' comparative running trials under similar conditions against a sister ship fitted with reciprocating engines showed a saving in coal consumption of 38·2 per cent in favour of S.S. *Mjölner*.

A somewhat larger vessel, S.S. *Wulsty Castle*, equipped on similar lines with two Ljungström turbines each of 625 Kw. capacity at 3600 r.p.m., built by the Brush Company in England, is provided with superheaters capable of raising the steam temperature to 625° F. A full description of this vessel's equipment has appeared in *Engineering*, Vols. CV and CVI, 1918.

A further notable advance in marine turbine equipment is marked by the S.S. *Pacific*, owned by the Overseas Company of Copenhagen, and built by the Copenhagen Dry Dock Company in 1920. In this case the reversibility is attained by means of a clutch, so that the loss in efficiency due to the

alternators and motors, which would necessarily amount to about 12 per cent, is reduced to that due to mechanical gearing, which is in the neighbourhood of 2 to 3 per cent.

The turbines as well as the gearing and reversible clutch were built by the Swedish Ljungström Company at their Finspong Works.

The vessel measures 375 ft. × 52 ft. × 35 ft. 8 in., and draws 24 ft. 6 in. with a displacement of approximately 10,300 tons, the block coefficient being about 0·76. The gross tonnage is 4088, and the net, 2505.

The following is a summary of approximately twelve months' performance as recorded in the chief engineer's log, and including voyages to and from Apia and Borneo partly on coal and partly on oil.

Total mileage	57,415.
Total hours	5610·5.
Mean speed	$10\frac{1}{3}$ knots.
Mean indicated horse-power ..	2100.
Mean coal per day	$22\frac{1}{2}$ tons.
Mean oil per day	$16\frac{1}{2}$ tons.
Coal per i.h.p. hour	1·00 lb.
Oil per i.h.p. hour	0·74 lb.
Coal coefficient	23,500.
Oil coefficient	32,000.

No stoppage at sea occurred during the twelve months due to main machinery. On opening up at end of year no perceptible wear was noticed on any part.

Fig. 52 shows the main turbine with its gearing and clutch. It will be seen that the turbine is of the standard Ljungström type, and has a capacity of 2100 i.h.p., its normal speed being 3000 r.p.m. The normal steam conditions are as follows. Steam pressure 180 lb. per square inch, steam temperature 630° F., vacuum 28 in. (Bar. 30 in.). The turbine discs are overhung on the two gear pinions, which revolve in opposite directions in accordance with usual Ljungström practice. One pinion engages directly with the intermediate speed gear wheel which runs at 540 r.p.m., but an idler is placed between the other pinion and the gear wheel, so that both halves of this gear wheel revolve in the same direction.

The reversing clutch is mounted on the shaft carrying this gear wheel, and is placed between this and the pinion of the second gear which drives the propeller shaft at 70 r.p.m.

The reversing clutch consists of a fixed outer casing, in which is mounted a revolving gear case containing a set of gear wheels of the "epicyclic" type. The gear case revolves with the intermediate shaft when the propeller is working ahead, and it remains fixed in the outer casing when it is going astern. These two conditions of the gear case are obtained by means of two sets of disc clutches, one inner and one outer. The mutual position of the outer disc is regulated by four pistons on which oil pressure can be applied on either side.

The inner set of discs is pressed together by means of oil pressure on a single large piston, and released by means of a spiral spring. This spring as well as the oil piston is placed concentrically with the gear case and the shafting.

The admission of oil to the pistons is regulated by means of a manœuvring wheel fitted near to the stop valve. When going ahead the outer discs are released and separated, and the inner ones are under pressure, causing the gear case to be fixed to the driving, and the driven parts of the case to be fixed

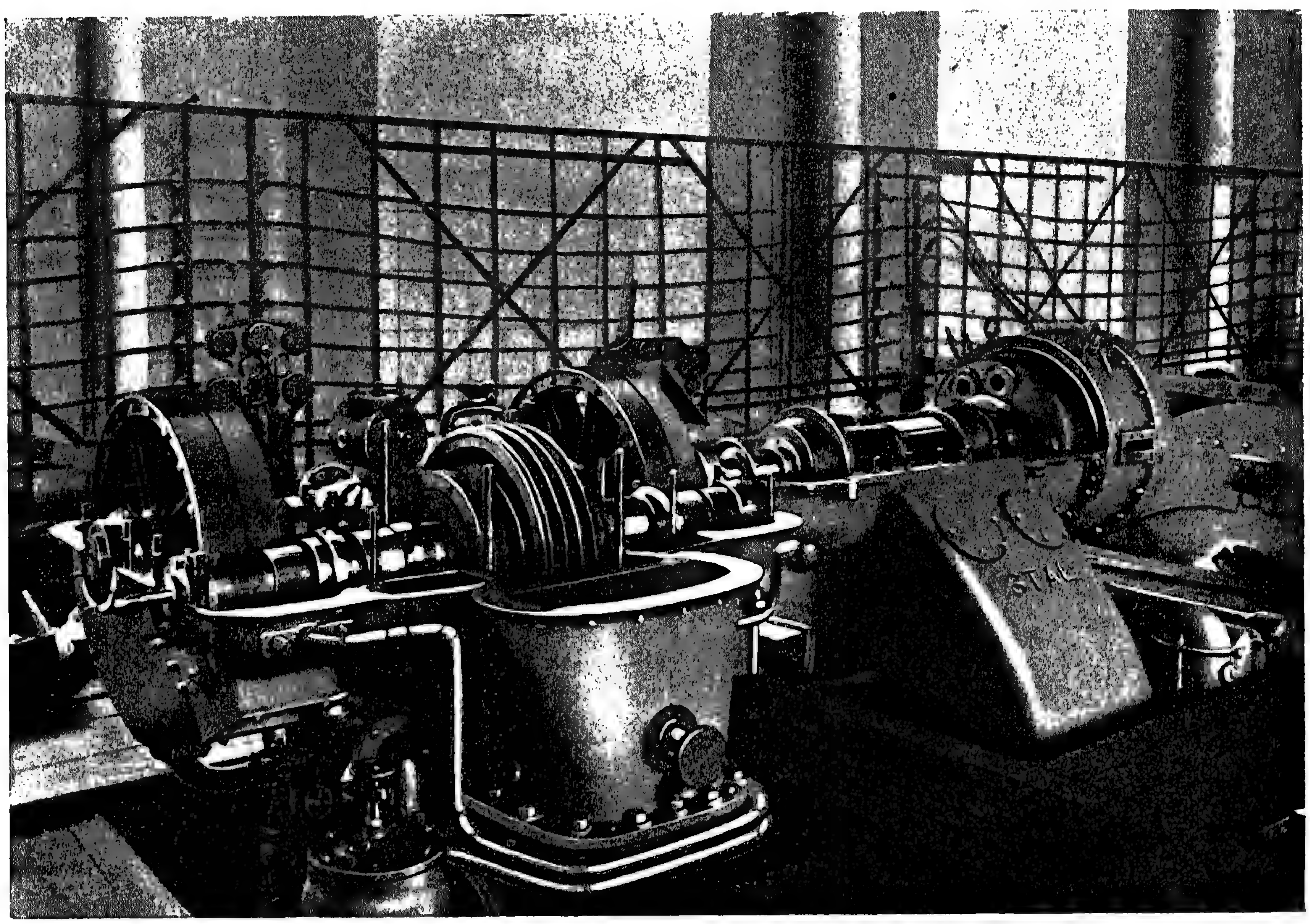


Fig. 52—Main Turbine of S.S. *Pacific*

both to the driving and the driven parts of the intermediate shaft. In this condition the gear case acts as a fly-wheel. The spur wheels inside the case remaining stationary in relation to the second reduction pinion; thus the reversing clutch causes no losses. When going astern the inner discs are released and the outer ones are pressed together. The gear case then becomes fixed relative to the outer casing, and the spur wheels inside the gear case are forced to revolve in opposite direction, thus also reversing the motion of the main shafting with the propeller. If both disc clutches are released, the propeller shafting and the second reduction gear become stationary, and the gear case rotates idly at half the number of revolutions of the intermediate shaft.

The supply of lubricating oil to the gear case is very ample, and even at the greatest possible continuous slipping of the clutches the oil is not likely to be overheated.

With this design of the reversing clutch there is obviously no necessity of stopping the turbine when manœuvring, but means are provided for a reduction of the speed down to 50 per cent, which is done automatically in connection with the manœuvring wheel mentioned above. A turn of this wheel in one direction or the other, more or less, is all that is required for manœuvring.

On the trial trip which was run in the end of December, 1920, the steamer had a displacement of 4130 tons, and with 1727 s.h.p. an average speed of 12·59 knots was attained with a consumption of 1·07 lb. of coal of 12,600 B.Th.U. per pound per s.h.p. hour. This consumption includes coal for all auxiliaries, steam steering gear, steam heating, and electric light. The corresponding net turbine consumption was estimated to be 0·912 lb. per indicated horse-power hour.

During the whole of the trial the machinery worked without a hitch, and the very satisfactory behaviour of the reversing clutch was particularly noted. The propeller was reversed from full speed ahead to full speed astern in 27 sec., and the clutch gear wheels worked so smoothly that it was impossible to tell from the sound which way the propeller was working.

Several additional vessels are being equipped on similar lines, and it is expected that the combination of high-speed turbines with mechanical gearing and reversing clutch will prove competitive in the matter of economy with the Diesel engine, which at the moment appears to hold the field for fuel economy in marine propulsion.

CONDENSERS AND COOLING TOWERS

BY

R. ROYDS, M.Sc., A.M.I.Mech.E.

Condensers and Cooling Towers

CHAPTER I

Condensers

General.—A condenser consists of a closed vessel of suitable form into which the exhaust steam from an engine is led and there condensed. The pressure of the steam entering the condenser depends primarily upon the temperature at which the steam condenses: the lower the temperature the lower is the pressure, as is seen by referring to the relation between the temperature and pressure of saturated steam given in the steam tables, Vol. IV, p. 225. To condense the steam the latent heat in the steam has to be absorbed by some cooling agent, usually cooling water supplied in sufficient quantity to take up the heat liberated, and the temperature at which condensation takes place is therefore largely dependent upon the temperature of this cooling water. For example, if the steam entering the condenser condenses at 110° F., a reference to steam tables shows that the pressure of the steam is 1·27 lb. per square inch absolute, or $(14{\cdot}7 - 1{\cdot}27) = 13{\cdot}43$ lb. per square inch below the normal pressure of the atmosphere. In this case the " vacuum " in the condenser would be $\frac{13{\cdot}43}{0{\cdot}491} = 27{\cdot}4$ in. of mercury. Therefore the back pressure on the engine piston would be correspondingly low during the exhaust stroke. When an engine exhausts to the atmosphere, however, the back pressure is slightly above atmospheric. It may be stated, therefore, that the main purpose of the condenser is to increase the amount of available work at the engine cylinder per pound of steam used compared with that obtainable with a non-condensing engine. In reciprocating steam-engine practice, however, it is generally considered that no appreciable increase in all-round economy results with condenser vacua higher than about 26 in. (barometer 30 in.). The main reason for this limit of economical vacuum is the inability of the reciprocating engine to expand the steam economically right down to the back pressure, the pressure at the point of exhaust in the low-pressure cylinder usually being several pounds per square inch above the back pressure.

The same arguments apply in a slightly modified form to the influence of the condenser on the amount of available work per pound of steam used by a steam turbine. But with steam turbines useful expansion can be obtained right down to the back pressure, and therefore high vacua can be used economically. For this reason condensers have undergone considerable developments in recent years, giving higher vacua and greater rates of condensation than formerly.

A certain amount of air or incondensable gases finds its way into the condenser under ordinary conditions of operation, and this necessitates an "air-pump" to withdraw this air as fast as it enters the condenser; otherwise the condenser would get full of air, the pressure would eventually rise to atmospheric pressure, and the condenser would then cease to be usefully operative. With reciprocating engines the air-pump is generally used also to extract the water as well as the air, in which case it is called a "wet" air-pump. In other cases the air-pump deals with the air only, and is then called a "dry" air-pump, a separate pump or its equivalent being used to extract the water.

The greater portion of the air which enters a condenser is due to leakage through the low-pressure piston rod or turbine glands, badly made exhaust-pipe joints, and porous castings. The feed water entering the boiler usually carries a small amount of air in solution, and this also eventually finds its way into the condenser with the steam. When the cooling or condensing water is injected into the condenser and comes into direct contact with the steam, as in jet condensers, the greater portion of the air which is in solution in this water comes out of solution in the condenser under the influence of the heating and the low pressure. Under ordinary circumstances, however, the leakage of air is a more serious factor than the air which comes out of solution from the water.

Every endeavour should be made to reduce the leakage of air into the condenser. Not only does the presence of air interfere with the condensation of the steam, but it also tends to increase the total pressure in the condenser, or, in other words, it tends to reduce the vacuum. When air and water-vapour are mixed together the total pressure exerted by the mixture is the sum of the pressures of the constituents, each exerted as if it occupied the space alone, and as if the other were not present. This pressure due to any constituent of a mixture is sometimes called its "partial pressure". Now the partial pressure of saturated steam is dependent only upon its temperature, whereas the partial pressure of air depends upon the weight of air present as well as upon the temperature. For example, if the total pressure at the air-pump suction is 2 lb. per square inch absolute, and the temperature of the mixture is 100° F., reference to steam tables shows that the pressure of steam at this temperature is 0·94 lb. per square inch. Therefore the partial air pressure is:

$$2 - 0{\cdot}94 = 1{\cdot}06 \text{ lb. per square inch.}$$

Again, the volume of 1 lb. saturated steam at 100° F. is about 350 c. ft.,

which also is the volume of the air associated with it. Now the volume V of 1 lb. air at 1·06 lb. per square inch pressure and 100° F. is:

$$V = \frac{53{\cdot}18 \times (100 + 460)}{1{\cdot}06 \times 144} = 195 \text{ c. ft.}$$

Therefore the weight of air associated with 1 lb. steam under these conditions is $\frac{350}{195} = 1{\cdot}79$ lb., and the ratio of the weight of air to the weight of steam is thus 1·79.

At the inlet to the condenser, however, the ratio of the air to the steam is extremely small, and assuming that the pressure there is also 2 lb. per square inch absolute, the steam would condense there at 126·2° F. Thus in this case the effect of the air in the condenser is that the steam condenses at the inlet at about 126·2° F., but near the air-pump suction at only 100° F. Had it been possible to exclude all air from the condenser, the temperature of the condensing steam all over the condenser at 2 lb. per square inch absolute would have been 126·2° F. The difference between the temperature of the steam or mixture at the condenser inlet and at the air-pump suction is, therefore, a rough measure of the amount of air entering the condenser. As is shown by the calculations on p. 237, however, the cooling and " devaporizing " of the air has an important influence on the capacity of the air-pumps.

Types of Condensing Plant.—Steam-condensing plant may be divided into two distinct types: (1) " Jet " condensers, where the steam and condensing water become intermixed, and (2) " surface " condensers, where the steam and cooling water are separated by a metal tube or plate. These may be further subdivided into " counter-current or contra-flow ", where the general flow of the steam is in an opposite direction from that of the water, and " parallel current ", where the flow of the steam and water is in the same direction.

The Jet Condenser.—The jet condenser is a relatively simple structure. In the common form usually adopted in reciprocating engines for mill and workshop driving, the condenser is a cast-iron chamber into which the steam exhausts, where it meets with the injection water sprayed into the condenser through a perforated pipe or rose. The water then falls to the bottom and is extracted along with the air by the wet air-pump driven directly by the engine. In fig. 1 is shown a jet condenser of the parallel-current type, as made by The Mirrlees Watson Co., Ltd., principally for use with steam turbines, the rotary air and water extraction pumps being independently driven. The exhaust steam from the turbine enters at the top, and the injection water enters directly beneath, issuing through a series of nozzles at high velocity in the form of spray, and thoroughly mixing with the steam. The mixed steam and water then pass through the cone, where the greater part of the steam is condensed. The cone is intended to allow the velocity energy of the water to compress slightly the air and incondensable

gases. This air is drawn from the condenser from under the cone by means of a rotary air-pump, and on the same spindle is a centrifugal pump for extracting the injection water, the spindle being driven either by an electric motor or a small steam turbine.

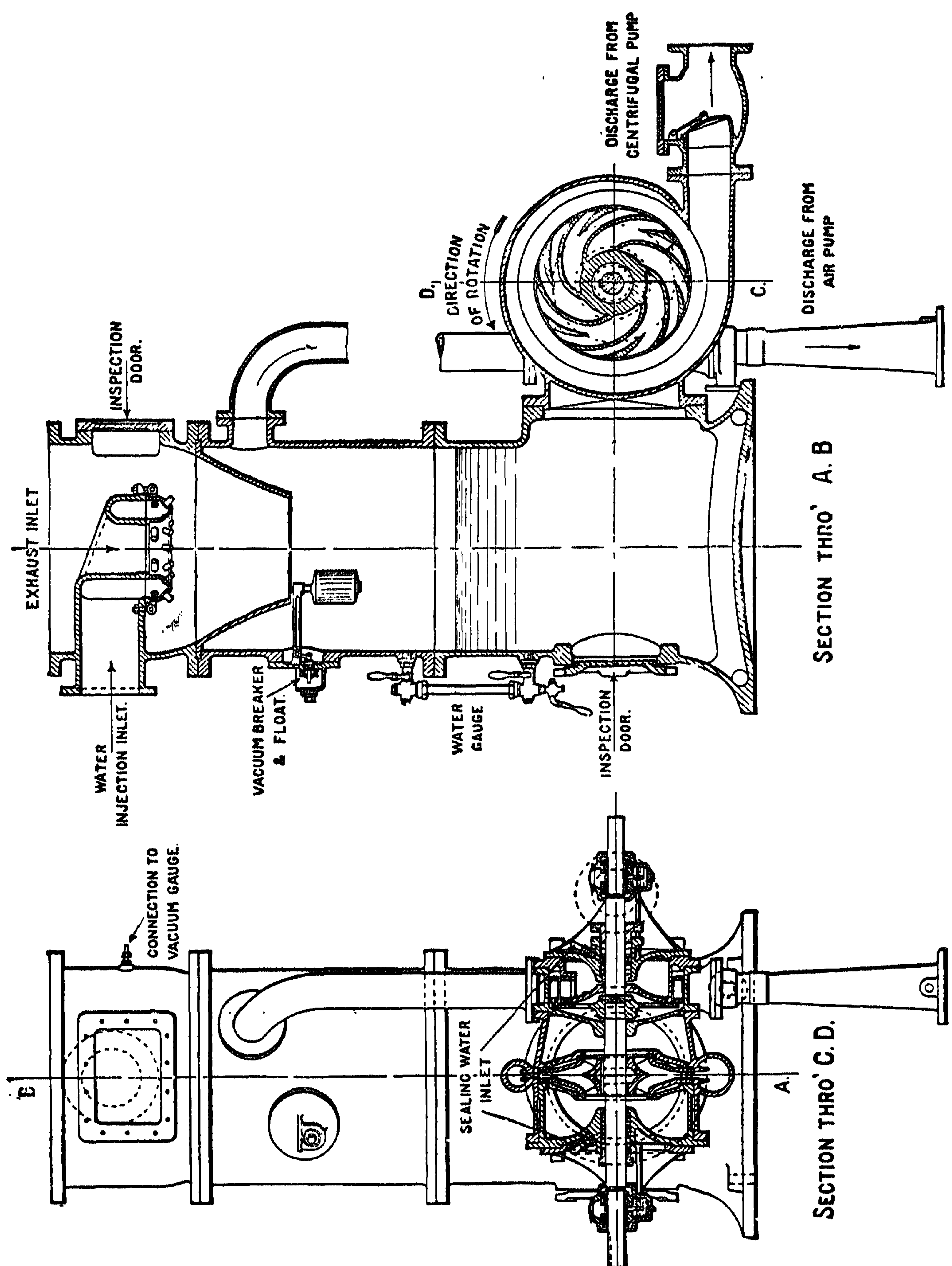


Fig. 1 —Parallel-current Jet Condenser with Rotary Pumps

One arrangement of the counter-current jet condenser as made by The Mirrlees Watson Co., Ltd., is shown in section in fig. 2, where there are fitted three distributing trays perforated with elongated holes about $\frac{3}{4}$ in. by $2\frac{1}{2}$ in., which serve to break up and distribute the condensing or injection water, and at the same time allow the use of fairly dirty water. If the injection water is very dirty the holes are dispensed with, and the trays are

made shallow to allow the water to fall over the edges in the form of a cascade, suitable openings being allowed in the cascades for the passage of the steam and gases. The air is drawn off at the top of the condenser after leaving the coldest condensing water, and is therefore of minimum volume. Where dry air-pumps are used it is advisable to fit a water separator in the air suction pipe, as shown in the figure. With this type of condenser the dis-

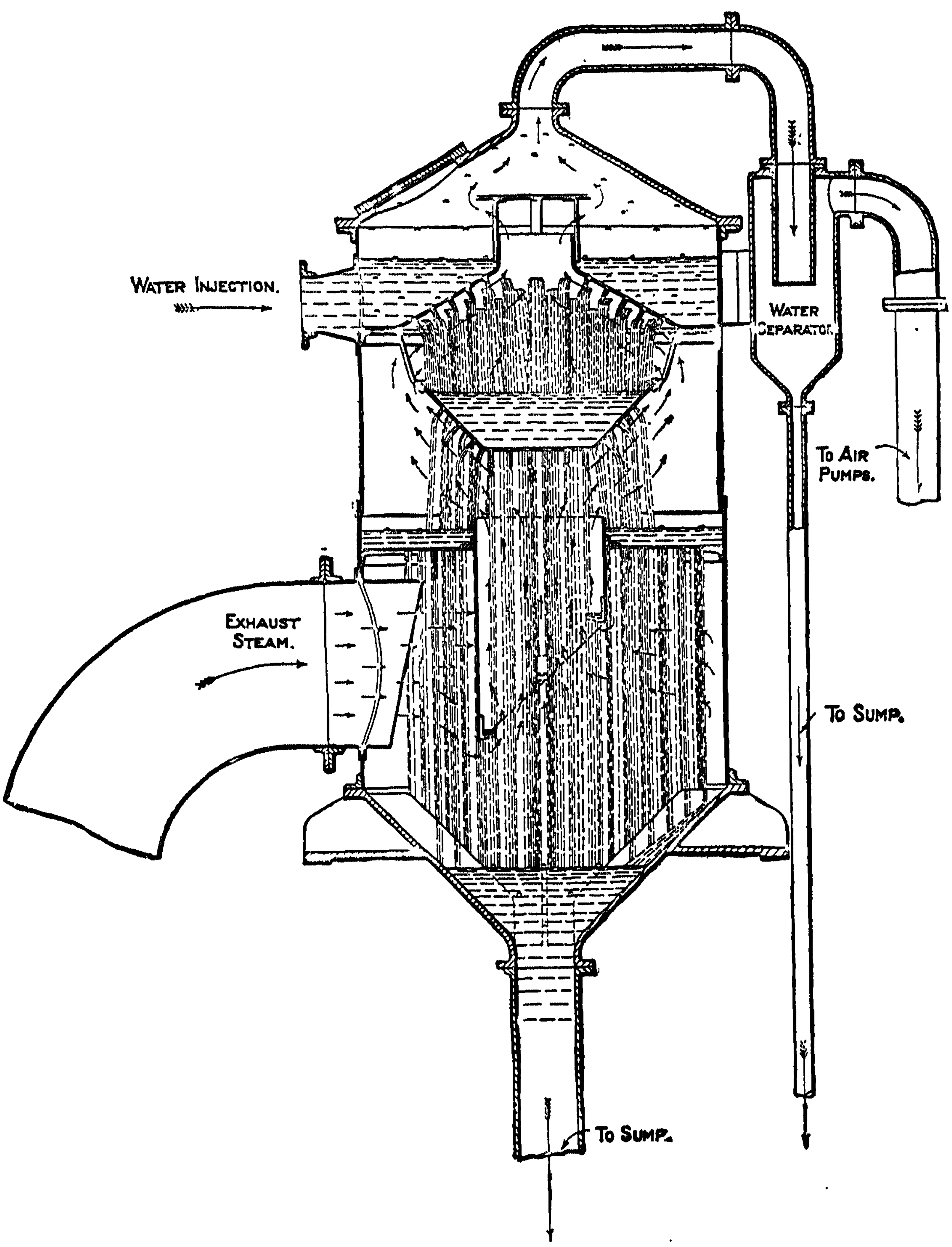


Fig. 2.—Counter-current Jet Condenser

charge temperature of the water at full load is only a few degrees below that of the entering exhaust steam.

The condenser shown in fig. 2 may be arranged at a low level with respect to the engine or turbine, or at a high level. In the low-level type the injection water is caused to flow into the condenser by the vacuum, but it is necessary to withdraw the water by means of a pump against a suction resistance equivalent to the vacuum, say against a head of 28 to 32 ft. of water.

There is a certain element of risk with this type, however, in the possibility of the water-extracting pump failing to work, causing the water to flood the condenser and possibly the main engine or turbine. A vacuum breaker (see fig. 1, p. 216) needs to be fitted to the condenser, arranged so that if by any reason the water rises in the condenser above a certain level a valve is opened automatically, admitting sufficient air to break the vacuum and cause the engine or turbine to exhaust to the atmosphere through a special exhaust relief valve on the exhaust main.

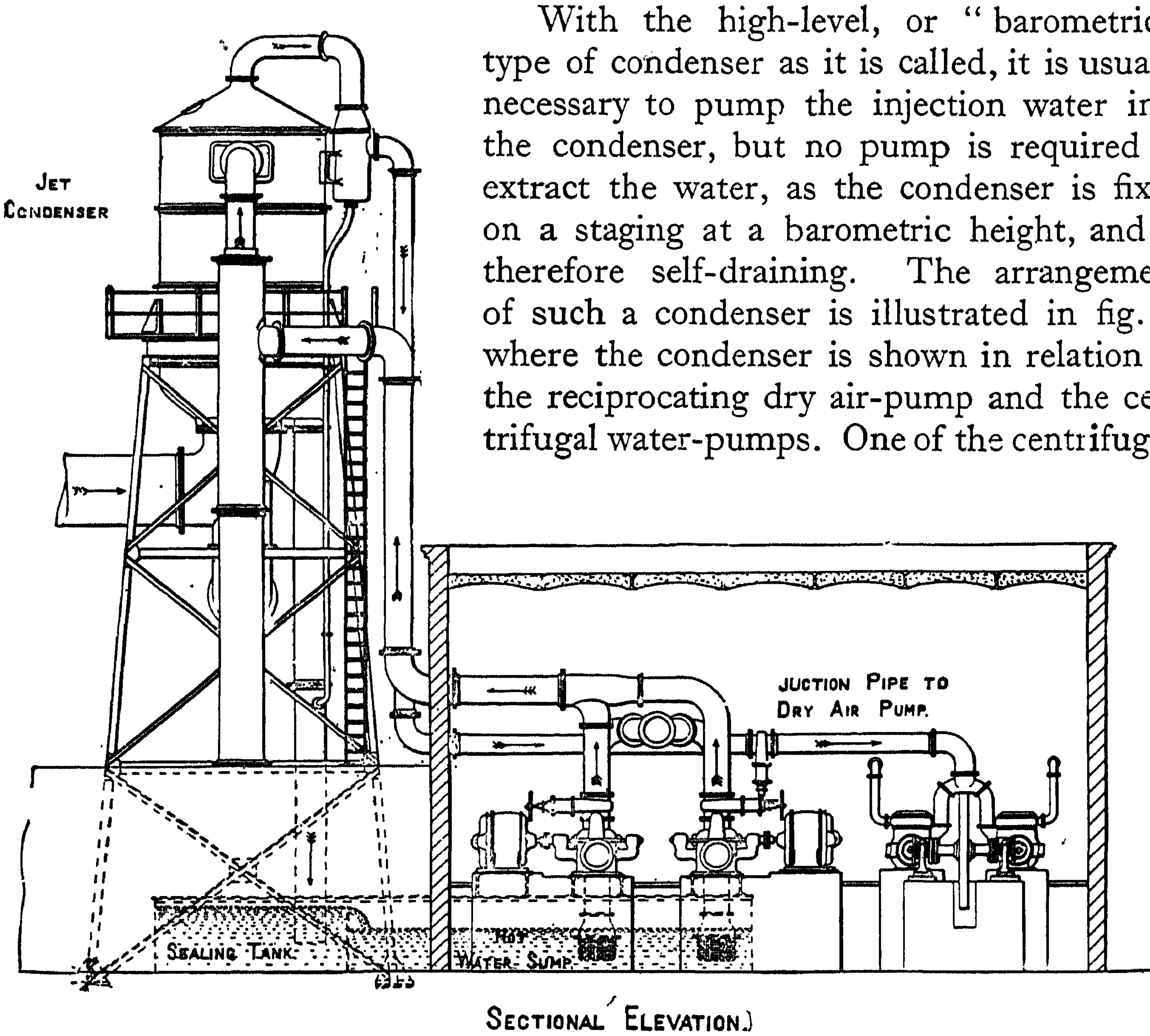


Fig. 3.—Arrangement of Barometric Condenser

With the high-level, or "barometric" type of condenser as it is called, it is usually necessary to pump the injection water into the condenser, but no pump is required to extract the water, as the condenser is fixed on a staging at a barometric height, and is therefore self-draining. The arrangement of such a condenser is illustrated in fig. 3, where the condenser is shown in relation to the reciprocating dry air-pump and the centrifugal water-pumps. One of the centrifugals delivers the water from the condenser-drain sump to a cooling tower, and the other delivers the water from the tower tank to the condenser. The vacuum in the condenser under ordinary conditions of operation is capable of lifting this water up the injection pipe to a height equivalent to the vacuum, and the pump is therefore only called upon to lift the water through the remaining height to the condenser inlet. But if the air-pumps are not independently driven there would be little or no vacuum in the condenser until water began to be supplied, and therefore this pump should be capable of giving the full lift at the start.

The barometric condenser has to be designed for a slightly higher vacuum than the low-level type, and there is more chance of air leakage, due to the

long exhaust-pipe. It is also more costly to install, because of the supporting staging and the long lengths of piping. Its main advantage, however, lies in its capacity for draining freely without risk of flooding, and there may also be a certain amount of saving of pumping power with the barometric type under favourable conditions.

What is known as the " ejector " condenser is illustrated in fig. 4, as made by Ledward & Beckett, Ltd. The injection water is supplied at the top with a head of about 15 ft., and acquires a considerable velocity in passing through the cone. If this head of water is not naturally available, usually a centrifugal pump is used to produce it. The exhaust steam passes through a large number of small inclined nozzles at a considerable velocity, and meets with the injection water flowing through the cone, and is thereby condensed. The momentum acquired by the mixture is sufficient to discharge the water against atmospheric pressure even when a high vacuum is produced in the exhaust steam connection. This requires an expanding nozzle or cone, as shown at the discharge end. Under suitable conditions the air which is mixed with the steam gets entangled in the water at the moment of condensation of the steam, and is discharged along with the water. So long as the amount of air to be dealt with is not abnormal the ejector condenser is capable of producing a fairly good vacuum, but with an unusual or abnormal air leakage the vacuum is apt to fall off considerably, and more or less erratic conditions are set up.

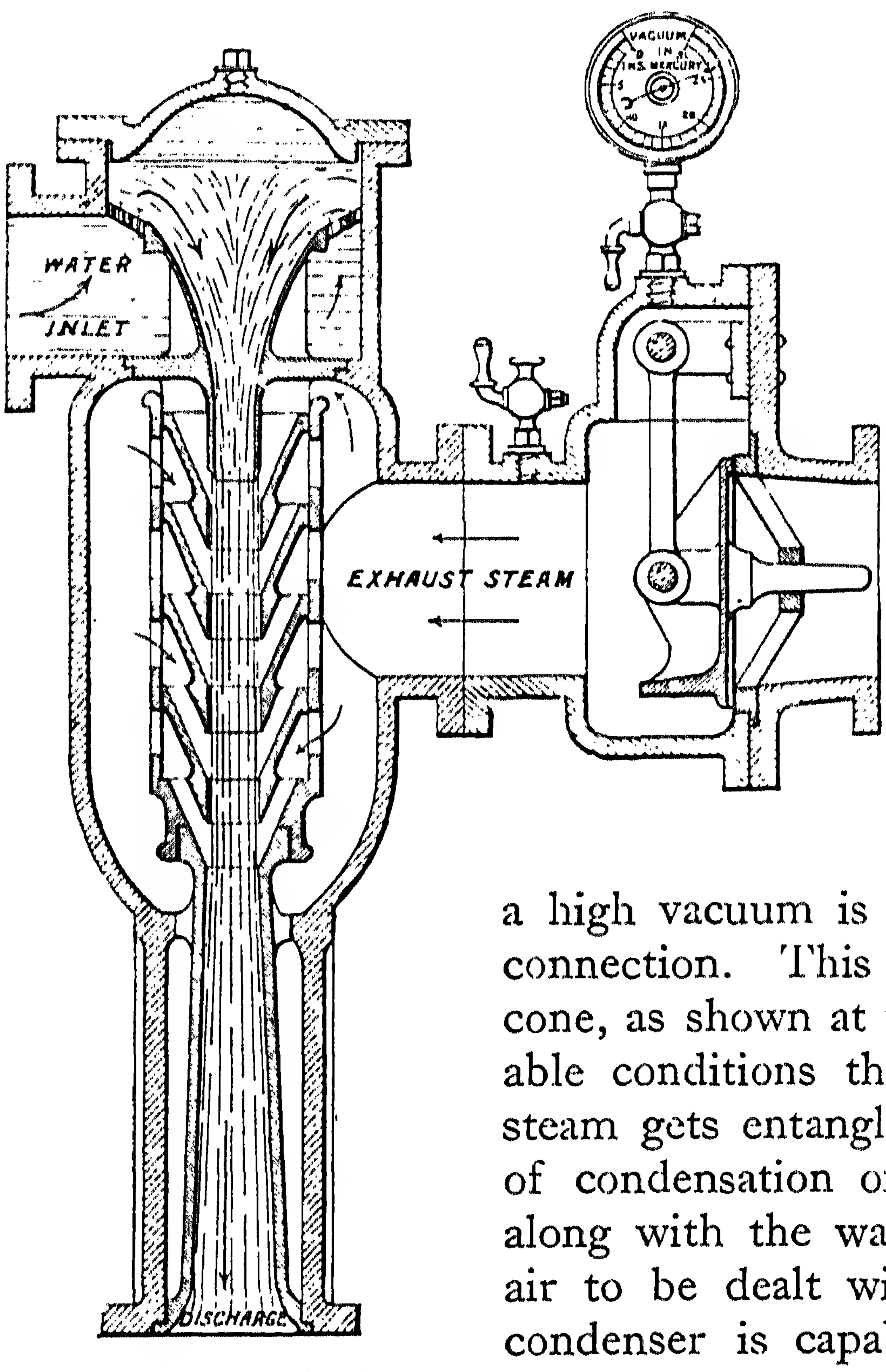


Fig. 4.—Ejector Condenser

The illustration in fig. 5 shows a modern arrangement (Westinghouse) of simple jet condenser, operating on lines similar in principle to the ejector condenser, except that the injection water passes through a Leblanc rotary pump. This type of pump is more particularly described on p. 242; but, briefly stated, the injection water in passing through the impeller is made to travel at a high velocity along the cones shown, carrying the air with it to the outlet, and compressing it sufficiently to discharge to the atmosphere. This type of pump is always arranged to lift the injection water, and therefore some arrangement is necessary for priming the pump with water. A steam starting ejector is arranged at C for this purpose, though if a supply of water

under pressure is available this steam ejector is not necessary. Such an arrangement of condenser is more certain in action than the ordinary ejector condenser shown in fig. 4.

The Rees Roturbo Manufacturing Co., Ltd., also build a jet condenser,

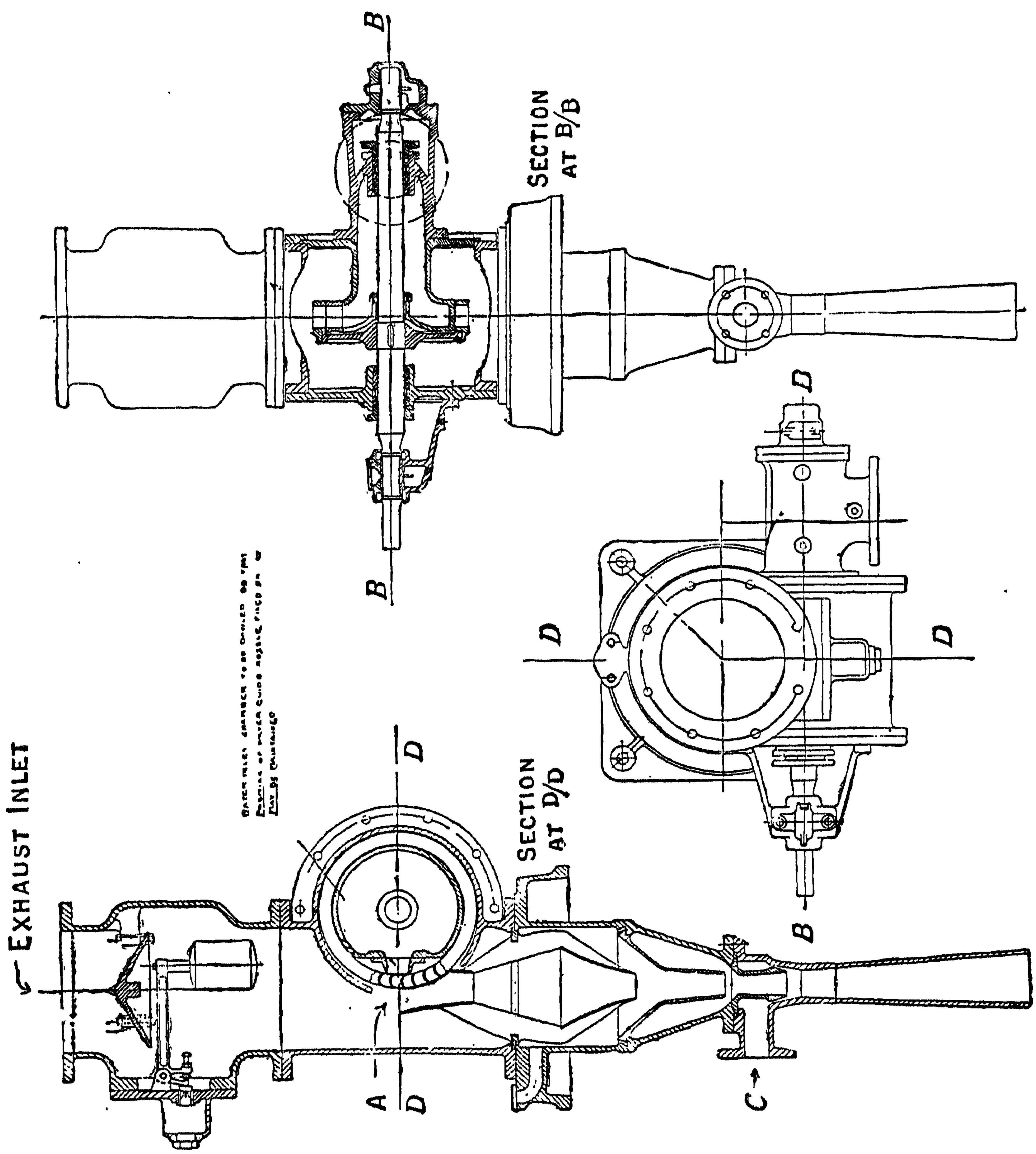


Fig. 5.—Ejector form of Condenser operated by Leblanc Rotary Pump

making use of their rotary air-pump or impeller to discharge the water and air, as illustrated in fig. 22, p. 243.

Condenser Calculations.—The most important calculation respecting jet condensers is the estimation of the amount of condensing water necessary. For this purpose it is necessary to know the amount of steam to be condensed, the inlet temperature of the condensing water, and the exhaust steam temperature required at the condenser inlet.

Let W = weight of injection water per unit of time.
w = ,, steam condensed ,, ,,
T_s = steam temperature, degrees Fahrenheit.
t_1 = inlet temperature of injection water, degrees Fahrenheit.
t_2 = outlet ,, ,, ,, ,, ,,
L = latent heat in each pound exhaust steam.

Then,
$$W(t_2 - t_1) = w(L + T_s - t_2),$$
$$\text{or } W = \frac{w(L + T_s - t_2)}{t_2 - t_1}.$$

If the heat in the steam supplied to the engine or turbine is known, and the work done by the engine or turbine is known per pound of steam used, it is possible to calculate the value of the latent heat L in the exhaust steam, making an allowance of, say, 5 per cent of the heat supplied to the engine for losses by radiation.

Thus, if H_s = heat in each pound of steam at the engine stop-valve, reckoned from water at exhaust temperature,
e = thermal efficiency of engine,
5 per cent = assumed losses by radiation, &c.,

then the latent heat L in each pound of exhaust steam is

$$H_s(1 - 0{\cdot}05 - e) = H_s(0{\cdot}95 - e).$$

For example, if the steam supply is superheated to 500° F., and the saturation temperature is 360° F., with exhaust temperature 120° F., and thermal efficiency 16 per cent,

Then, from steam tables, H_s = 1183 B.Th.U. per pound (from water at 120° F.),
and $L = 1183\,(0{\cdot}95 - 0{\cdot}16)$,
$= 935$ B.Th.U. per pound.

This is based on the assumption that practically all the steam used by the engine passes to the condenser, which, of course, is usually the case in practice.

For approximate calculations it is commonly assumed that 1 lb. exhaust steam carries about 1000 B.Th.U. to the condenser as latent heat.

Continuing this example, suppose the injection-water inlet temperature t_1 is 76° F., and the outlet temperature of the mixture is 114° F.

$$\text{Then } W(114 - 76) = w(935 + 120 - 114),$$
$$\text{or } \frac{W}{w} = \frac{941}{38}$$
$$= 24{\cdot}8 \text{ lb. water per pound steam condensed.}$$

There are other factors, however, in the working and design of jet condensers which have to be carefully considered. The supply of injection water is usually at a lower level than the point of injection, so as to remove the principal danger of the condenser and engine becoming accidentally flooded with water. The vacuum produced by the air-pumps is depended upon to inject the water, and therefore there are practical limits to the height the water may be raised by this means. The lift is not usually made more than 12 or 14 ft., but in special circumstances may be as much as 20 ft. In addition there is the resistance to the flow of the water through the pipes and injection valve, and this can be estimated approximately when particulars of the piping are available, but this resistance may be roughly taken to be about 4 ft. of water. Thus, if the vacuum is 27 in. of mercury, lift of water 12 ft., and pipe resistance 4 ft. of water, the head of water available for injection would be

$$\frac{27}{12} \times 13{\cdot}6 - 12 - 4 = 14{\cdot}6 \text{ ft. of water,}$$

where 13·6 is the specific gravity of mercury.

If each injection nozzle has a rounded entrance, the coefficient of discharge is nearly unity, and thus:

$$\frac{v^2}{2g} = 14{\cdot}6,$$

where v is the maximum velocity of injection obtainable under these conditions, or

$$v = \sqrt{2 \times 32{\cdot}2 \times 14{\cdot}6} = 30{\cdot}7 \text{ ft. per second.}$$

If the steam condensed per hour is, say, 10,000 lb., and $\frac{W}{w} = 24{\cdot}8$ from the previous calculation, then

$$\begin{aligned} W &= 24{\cdot}8 \times 10{,}000 = 248{,}000 \text{ lb. water per hour,} \\ \text{or, } &= \frac{248{,}000}{62} \\ &= 4000 \text{ c. ft. per hour.} \end{aligned}$$

Thus, if a is the area of the nozzles at the throat in square feet,

$$\begin{aligned} a \times 30{\cdot}7 &= \frac{4000}{3600}, \\ \text{or, } a &= 0{\cdot}0362 \text{ sq. ft.} \\ &= 5{\cdot}21 \text{ sq. in.} \end{aligned}$$

The size of each nozzle or orifice would depend to some extent upon the amount of foreign matter which might pass the " snore " pipe or strainer in the intake. If $\frac{1}{4}$ in. diameter were adopted, the minimum number would be given by

$$\begin{aligned} n \times 0{\cdot}25^2 \times \frac{\pi}{4} &= 5{\cdot}21, \\ \text{or, } n &= 106. \end{aligned}$$

There is always some possibility of some of the nozzles becoming choked with dirt, fibrous matter, small leaves, weeds, &c., especially when the water is taken from a river. It is therefore the usual practice to increase the number of nozzles to allow for this, and to throttle the water-supply somewhat at the injection valve for normal working. Similar provision may also be necessary for overloads on the engine according to circumstances.

If for any reason the vacuum in the condenser should fall off, as might occur, for instance, if the air-pump became defective, this would correspondingly affect the head of water available for injection, and if the injection valve could not be opened further, the amount of water injected would also fall off and the vacuum thereby would be further reduced. Thus there will be some critical vacuum below which the amount of water available is not sufficient to condense the steam, and the vacuum would then fall rapidly towards zero. Calculations indicate that this critical vacuum is not far from the vacuum corresponding to the suction lift under ordinary conditions of operation, and this illustrates one of the dangers of having an excessive suction lift.

When the air-pump is driven directly from the engine, and there are no auxiliary means of creating a vacuum in the condenser, the condenser pressure will usually be atmospheric on starting the engine. The air-pump is depended upon to build up the vacuum to the point at which water begins to be injected, the steam exhausted during this period of no-load running being condensed by the relatively cold metal and water present in the condenser. It is therefore necessary to have an efficient air-pump of large capacity, and to keep down the designed volume of the jet condenser to a safe value, or the vacuum required to lift the injection water may never be reached, and the condenser therefore fails to get water. In such cases the condenser volume is usually made about one-half that of the low-pressure cylinder, and the volumetric displacement of the air-pump per stroke about one-third the volume of the condenser.

With independently-driven air-pumps, however, the pumps are started before the engine or turbine, and therefore this arrangement has greater stability of operation than that of direct-driven air-pumps.

Surface Condensers.—When suitable feed water is not available for the boilers, or is too costly to use for this purpose, it is usual to install surface condensers. The surface condenser most commonly used consists of a closed vessel of suitable shape containing tubes, usually of brass, on which the steam condenses. The cooling or circulating water is circulated through the tubes and absorbs the heat from the steam; thus the water of condensation is kept separate from the circulating water, and is therefore available for feeding again to the boilers.

One arrangement of a counter-current surface condenser * (Westinghouse) is shown in fig. 6. The tubes are supported in the two tube plates in a manner similar to that shown in fig. 11, p. 228, and are packed so as to prevent leakage of circulating water to the steam side of the tubes. The circulating

* *Proceedings of the Institution of Mechanical Engineers*, January–February, 1913.

water enters the end cover at the point D, and flows through the bottom half of the tubes to the other end. At this end the cover directs the water through the upper half or top set of the tubes, and it finally leaves the condenser at the point E. The exhaust steam enters at A through exhaust connections of ample area, which is particularly necessary for steam turbines expanding to very low pressures, and passes among the tubes. Some of the tubes in the top set are left out to form channels so as to facilitate the proper distribution of the steam over the whole cross-section of the condenser. Near the bottom of the condenser the baffle F directs the mixture of air and the small amount of uncondensed steam to the air-pump suction C, whilst the water of condensation is extracted at B. The tubes behind the baffle F between the bottom of the condenser and the air-pump suction C are introduced to cool and "devaporize" the air. This reduces the volume of the air and the corresponding displacement of the dry air-pump as is shown by the calculations on p. 237. The devaporizing tubes are usually adopted for high vacua and therefore are hardly necessary in condensers connected to reciprocating engines.

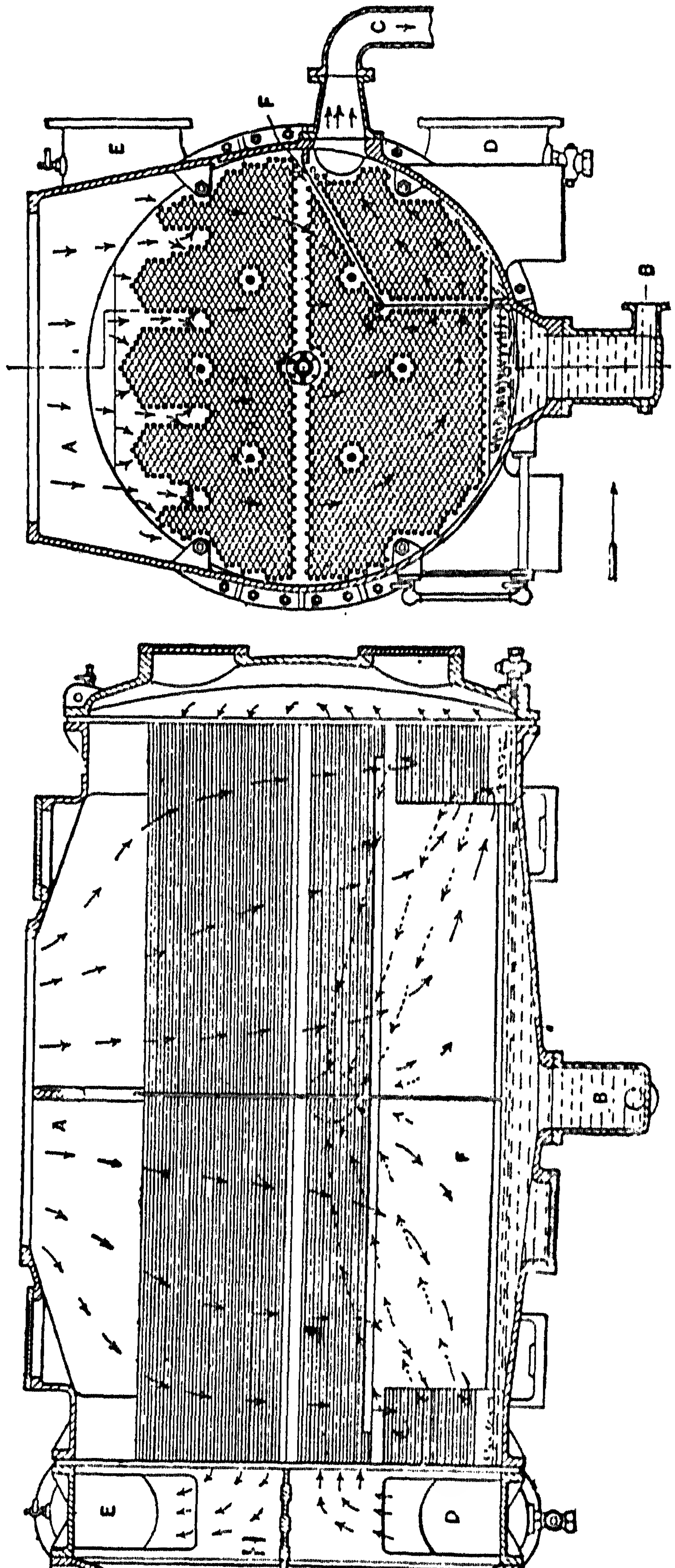


Fig. 6.—Surface Condenser

An inspection of the cross-sectional view in fig. 6 shows how the area available for the flow of the steam and air narrows down from the steam inlet to the air outlet. This is a desirable feature, so as to preserve a reasonable velocity of flow of the steam over the tubes as the condensation goes on. With a circular shape of shell, however, this feature cannot be arranged for as well as in a condenser of

heart-shaped or triangular cross-section, but the circular shape is not so costly to manufacture, and it is more easily stayed against the collapsing pressure of the atmosphere than the other shapes mentioned.

The illustration in fig. 7 represents the arrangement of tubes and baffles adopted in the circular condenser by the Contraflo Condenser and Kinetic Air-pump Co., Ltd. In order to reduce the resistance to the flow of the steam to the lowest value, the exhaust steam is allowed to have access to the tubes round the greater portion of the circumference. To prevent short-circuiting and stagnation, baffles are introduced, as shown, which are arranged to reduce the cross-sectional area available for the flow of the steam and air gradually towards the air-pump suction. It is also claimed that the baffles ensure the highest possible temperature for the water of condensation by allowing it to drain off to the bottom of the condenser, instead of dripping over all the lower tubes. The water of condensation is withdrawn at the central connection at the bottom of the condenser, while the air and remaining vapour passes underneath the lowest baffle across the devaporizing tubes to the side outlets under this baffle.

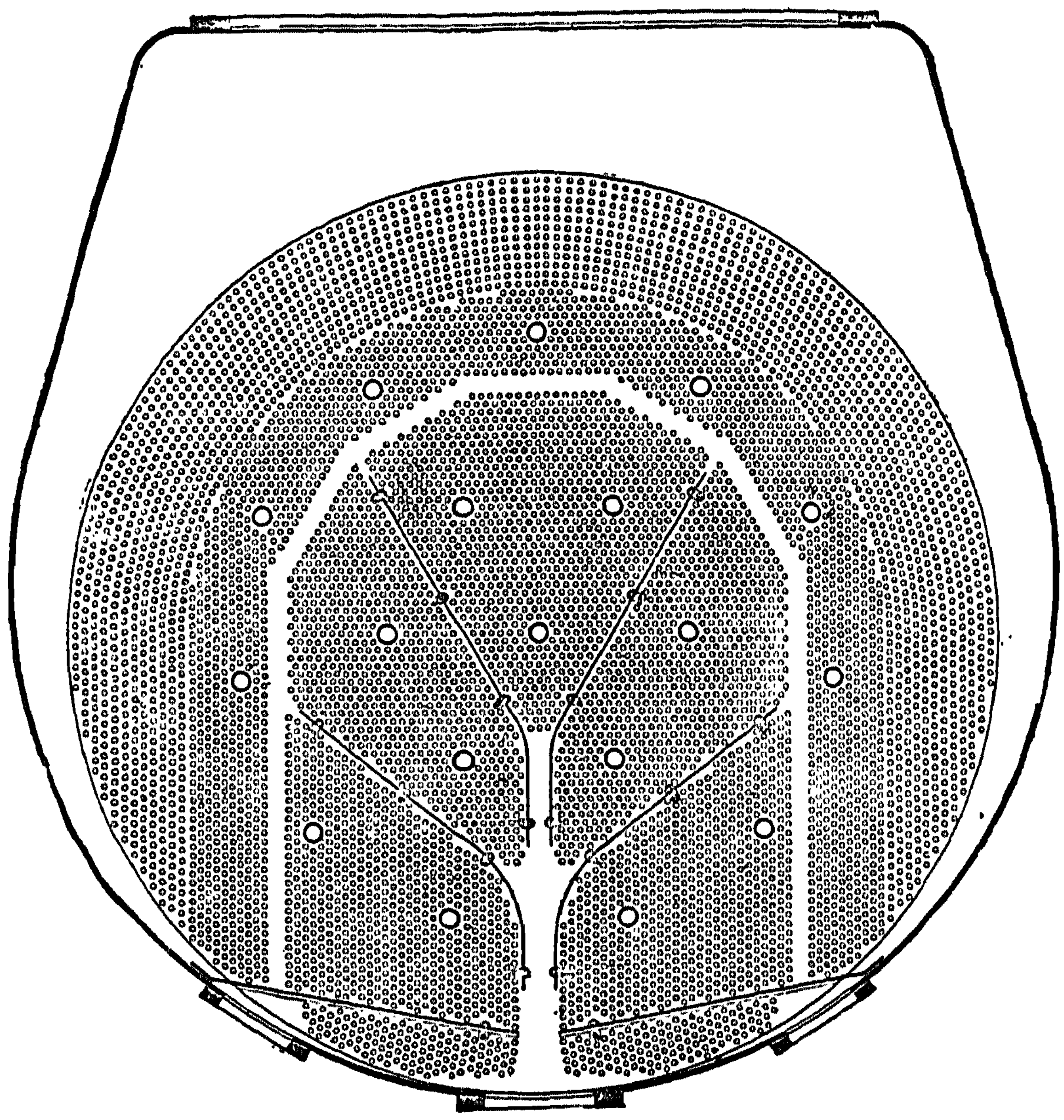


Fig. 7.—Surface Condenser with Directing Baffles

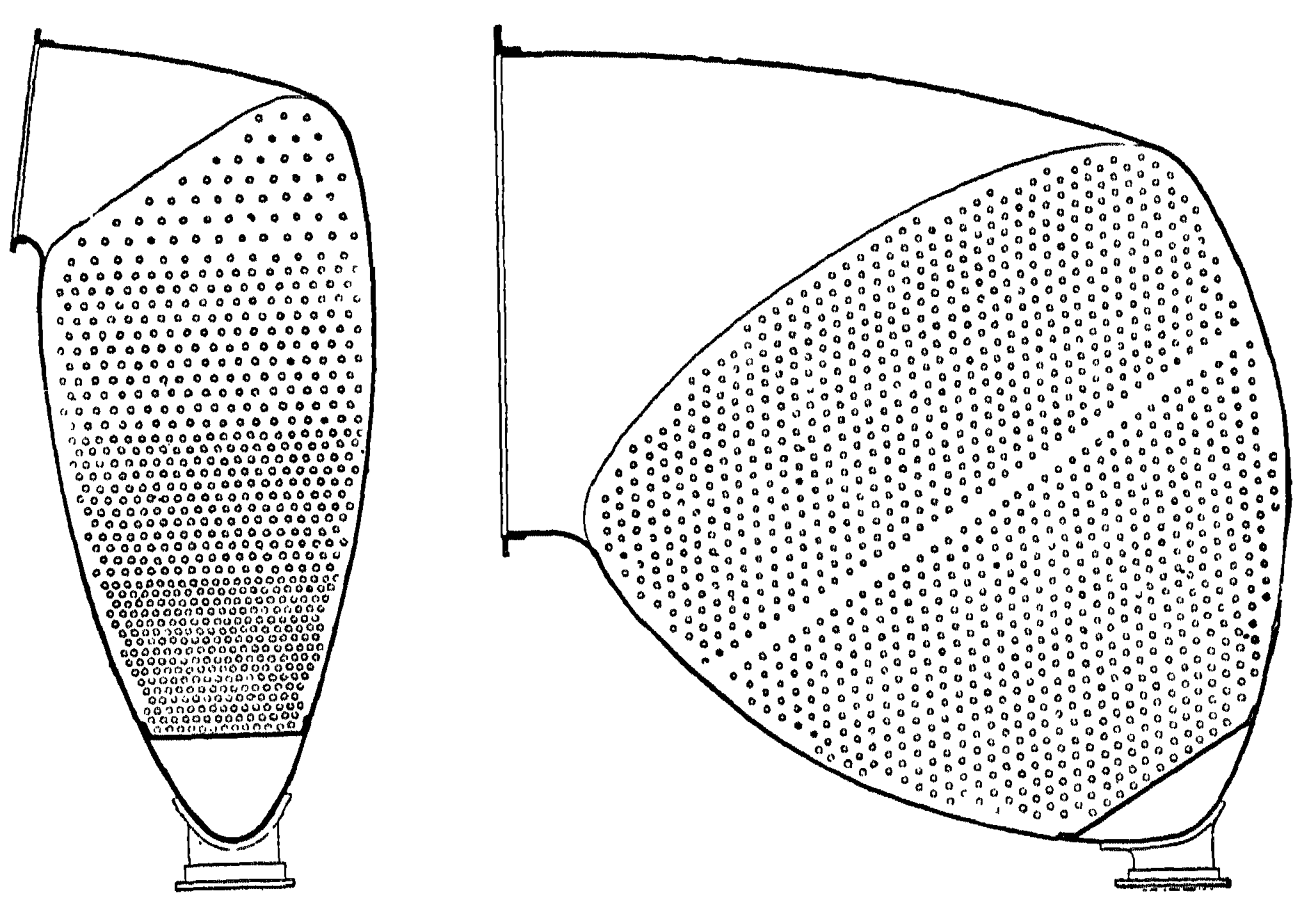


Fig. 8.—Tube Plate Diagrams

Fig. 8 shows representative tube-plate diagrams of the "Uniflux"

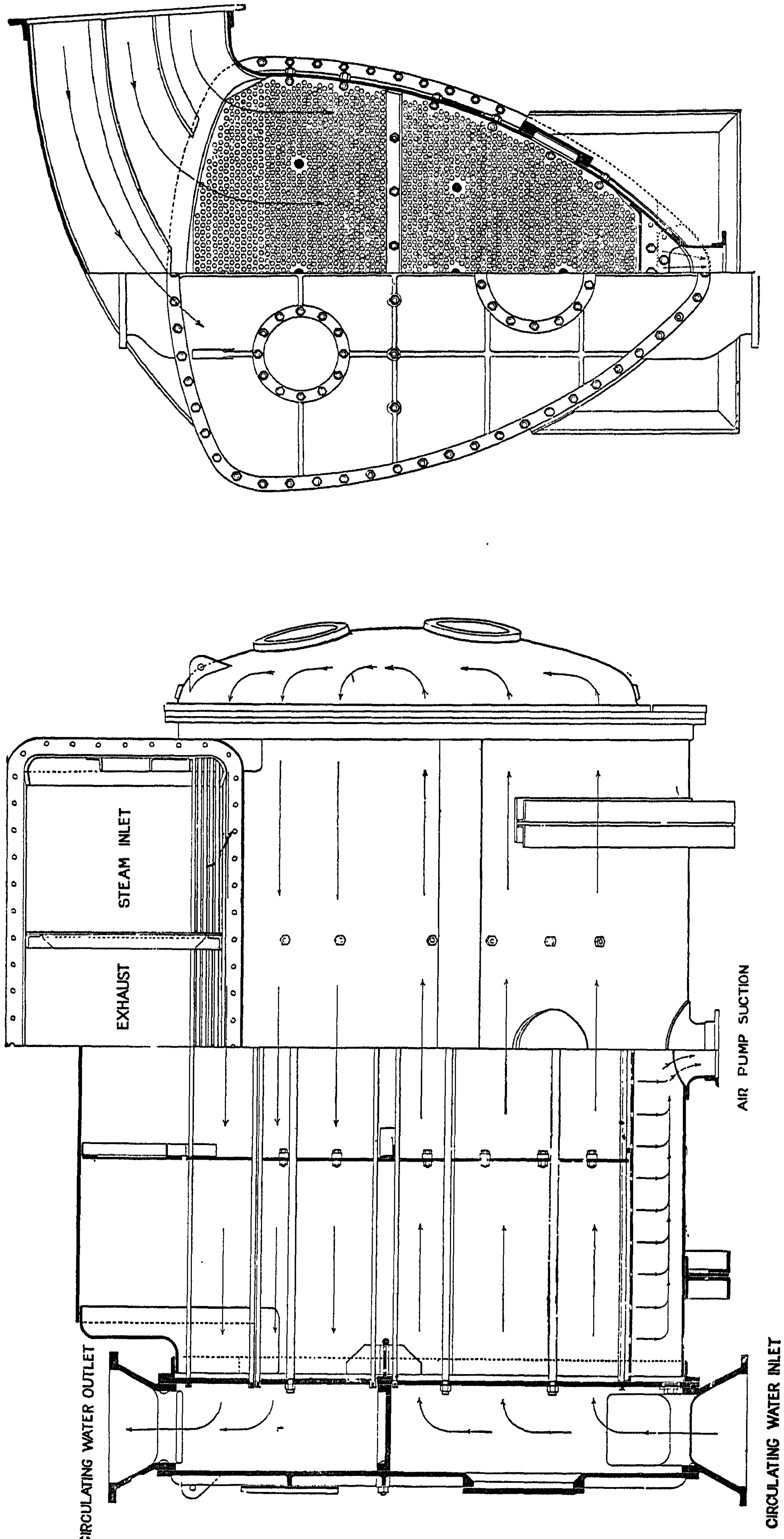


Fig. 9.—Weir "Uniflux" Condenser

condenser built by Messrs. G. & J. Weir, Ltd., and it will be noticed that they are more or less of heart-shaped section. The area available for the flow of the steam therefore gradually decreases as the steam condenses, and this is also affected by the closer spacing of the tubes towards the bottom. The air and its associated vapour along with the water of condensation are withdrawn at the bottom or narrow end of the condenser.

In fig. 9 are shown two sectional views of a Weir "Uniflux" condenser arranged for connection to a marine turbine where the condenser stands beside the turbine. Between the lowest tubes and the bottom of the condenser a perforated plate is introduced, the perforations being numerous to prevent any appreciable resistance to the flow at this point, but at the same time induce the stream of fluid to distribute itself over the length of the condenser instead of becoming localized near the air-pump suction. In this manner, then, stagnation on the steam side of the lower tubes is prevented. Usually the "dual" air-pump shown in fig. 15 would be connected up to the bottom of the condenser.

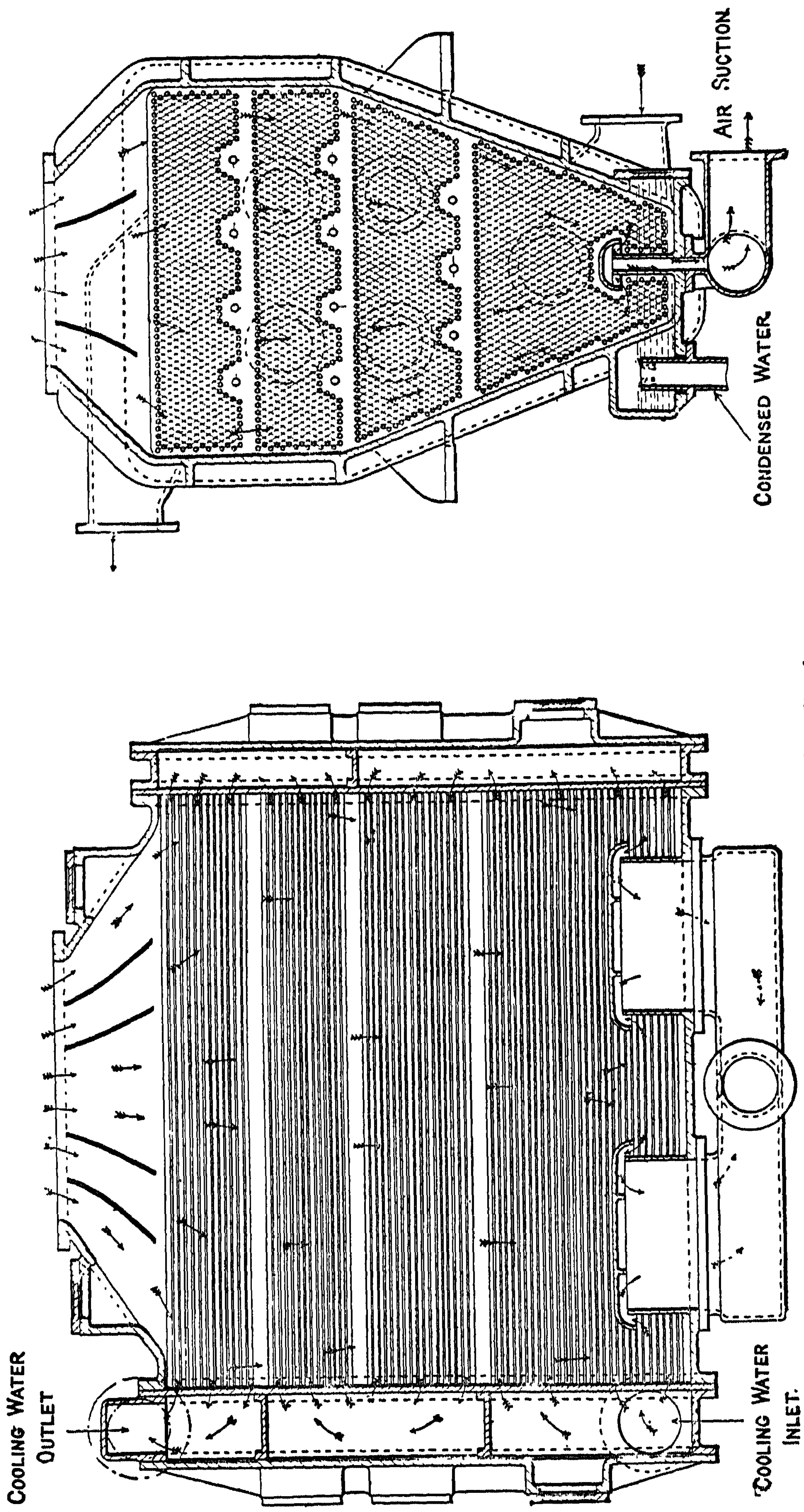


Fig. 10.—Surface Condenser

The surface condenser shown in fig. 10, built by The Mirrlees Watson Co., Ltd., approximates to the heart-shape section, deviating slightly from it for reasons of manufacture. It is readily seen that the velocity of flow

across the tubes is kept more nearly constant than it is, for instance, in a condenser of rectangular cross-section, and thus stagnation among the tubes is largely obviated. The air and water of condensation are extracted at different points, and a few of the bottom tubes are allowed to flood, so as to cool the water somewhat before passing to the pump. This makes the water-extraction pump more certain in action, but in general it is hardly desirable to cool the water of condensation when it is to be pumped back into the boiler, because of the loss of heat entailed by cooling the feed unnecessarily. When the condenser is designed for high vacua in connection with steam turbines, flooding of the lower tubes would be dispensed with, and a baffle introduced to cut off some of the tubes for devaporizing the air on its way to the air-pump suction. It is seen from the longitudinal section in fig. 10 that the water is here arranged to make four passes from the inlet to the outlet. This arrangement may be modified, of course, according to circumstances, but, all other things equal, the greater the number of passes the shorter is the condenser and the greater is the number of tubes. A condenser of this shape does not need to be placed in the vertical position. To save head room it can easily be arranged with one of the inclined sides on the foundations, the condenser then lying on its side.

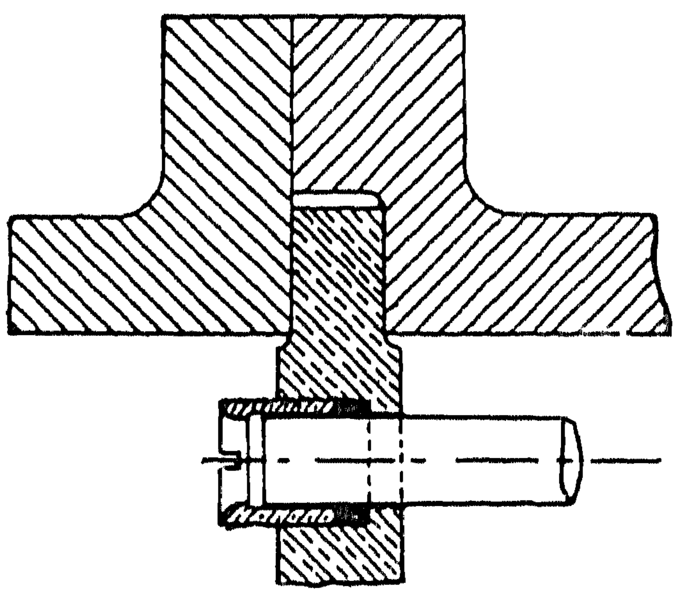


Fig. 11.—Attachment of Condenser Tube and Tube-plate

In regard to the best diameter of tubes to be employed, this depends to a large extent on the conditions of the water-supply. The smaller the diameter of the tube the more effective is the surface from a heat transmission point of view, but few makers care to insert tubes less than $\frac{5}{8}$ in. outside diameter (thickness No. 18 W.G., or 0·049 in.), even when the circulating water is quite clean. Three-quarters of an inch is the standard used by most manufacturers, but where the water is more or less dirty, and therefore liable to choke the tubes, it may be necessary to adopt tubes of 1 in. diameter.

The usual arrangement of packing, and the gland ferrule for the ends of the tubes, is shown in fig. 11. The tube-plate is bored slightly larger than the diameter of the tube, and is also screwed to accommodate the ferrule. To make the gland water-tight, soft yarn treated with linseed-oil or tallow is inserted, and the ferrule screwed down on it. It is advisable to have the inside edge of the ferrule rounded off, as shown, to reduce the loss of energy of flow at the inlet to or outlet from the tubes. Until recently it was the common practice to bolt the tube-plates to the ends of the condenser shell, and with the end covers bolted up against the tube-plates, as is shown in fig. 6, p. 224. There is thus a made joint between each tube-plate and the shell which it is impossible to re-make without detaching the tube-plate and the tubes. A more modern arrangement is that shown in fig. 11, where the joint subject to the vacuum is between the cover and the shell, and can be

re-made by detaching the cover without disturbing the tubes or the tube-plate.

Circulating pumps are usually of the centrifugal type driven by electric motor, steam turbine, or high-speed reciprocating engine, according to their particular convenience. When the circulating water is taken from a river, canal, or cooling pond, the pipes are generally arranged on the closed or siphon system, that is, both the suction and delivery ends of the pipes are placed below the surface level of the water. In this case the power required for pumping is only that due to the resistance to the flow through the pipes and condenser. The over-all efficiency of a centrifugal pump when driven by an electric motor may be taken at 40 to 50 per cent if working freely without throttling of the water-supply.

In large-power installations the intake from the river should be well protected from the entry of weeds, leaves, &c., by means of a screen. This sometimes requires constant cleaning to prevent choking of the screen. To obviate this, self-cleaning rotary screens have been devised, such as that built by Messrs. Ledward & Beckett, Ltd.

In connection with the steam plant for a large building, works, or mill requiring a large amount of steam for heating purposes, the steam-engines or turbines may be arranged to exhaust into the pipes of the heating system, either at a pressure of 5 to 10 lb. per square inch above the atmosphere if the engines run under non-condensing conditions, or with a moderate degree of vacuum if running condensing. The heat in the exhaust steam may then be usefully employed for heating purposes, and thus save a corresponding amount of boiler steam. When the steam plant is shut down, then boiler steam would be supplied to the heaters if required. In summer the engines might exhaust into an ordinary condenser in the usual way if the exhaust steam is only required for heating the building.

As to whether it pays to run a steam plant in this manner depends largely upon the circumstances of each case, that is, as to whether the steam plant uses a large or a small amount of steam, and whether an appreciable amount of steam is required for heating purposes when the plant is running.

Rate of Heat Transmission in Surface Condensers.—Since the advent of the steam turbine, and because of its ability to use efficiently the highest vacuum obtainable in the condenser, much greater attention has been paid to the factors which influence the rate of heat transmission in surface condensers. The most important of these factors are the reduction of air leakage into the system to the lowest possible limits, the influence of the velocity of the water passing through the tubes, and the cleanliness of the tube surfaces. To prevent leakage of air as much as possible, the turbine glands are usually steam-packed, so that any leakage of the gland is that of steam into the turbine casing. Also, when the condenser plant is erected, the condenser and the exhaust-pipe connections should be blanked off and filled with water, which is then subjected to a pressure of a few pounds per square inch above atmospheric pressure. If any of the joints are leaky it then becomes evident by the leakage of the water; but notwithstanding all

precautions it is practically impossible to prevent some leakage of air when the condenser is in operation. Another method of testing for air leakage is to run the air-pumps to obtain a vacuum in the condenser, and after stopping these pumps note the fall of vacuum occurring. If the system is reasonably tight, the vacuum should not fall more than a small fraction of an inch of mercury in, say, twenty or thirty minutes. It is sometimes found that the leakage of air into the condenser increases gradually in course of time, with a consequent reduction of the vacuum unless the air-pumps are of ample capacity. On occasion a joint may give way somewhat suddenly and allow quite an abnormal air leakage. Under the pressure of ordinary working conditions the engineer in charge may not notice or may not recognize that these leakages are occurring, and may be ascribing the faulty vacuum to other incidental circumstances.

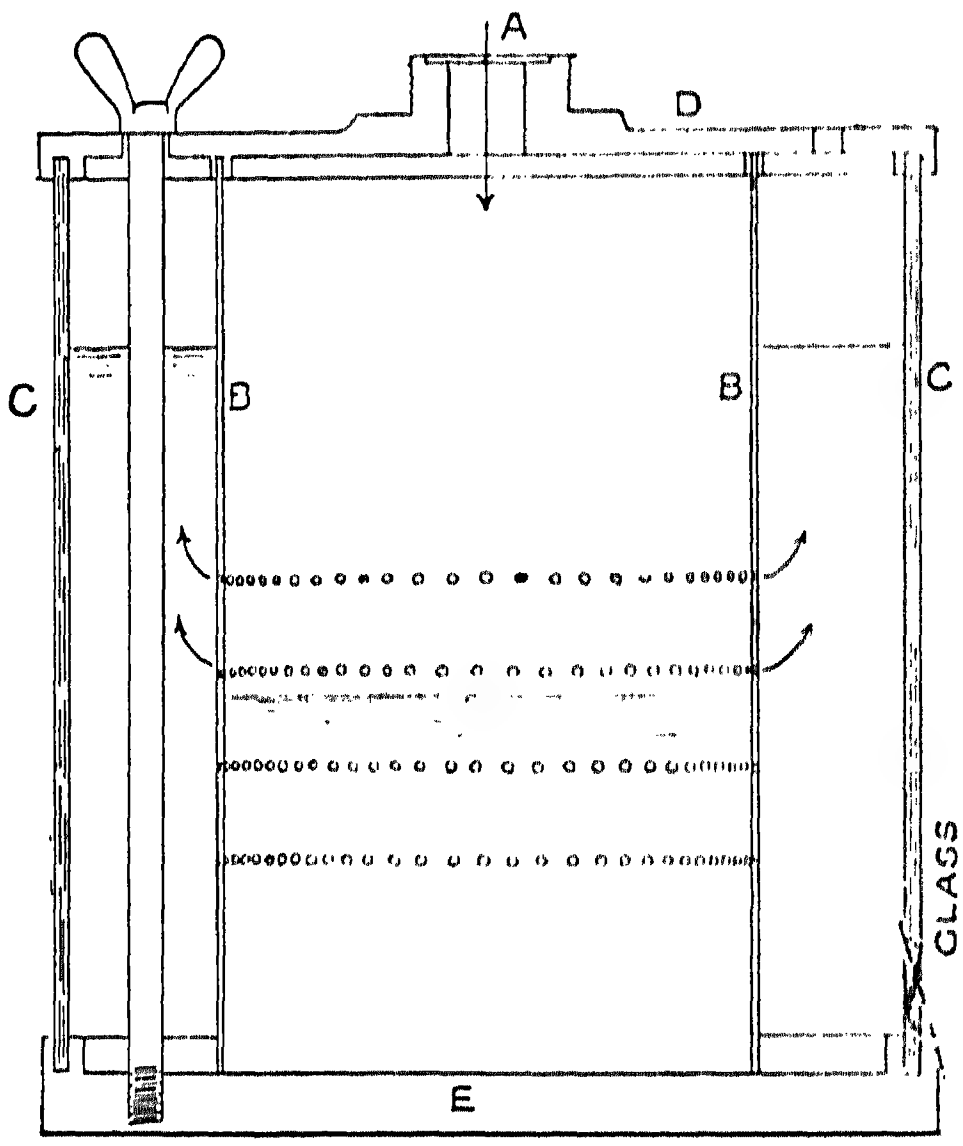


Fig. 12.—Weighton Air Indicator

In order to indicate leakage of air, Professor Weighton has designed an ingenious air indicator, one form of which is illustrated in fig. 12. Instead of allowing the air to be discharged to the atmosphere by the air-pumps, connection is made to the indicator at A so that the air is delivered inside the vessel or bell B, and therefore it depresses the water-level sufficiently to allow the air to escape to the atmosphere through the water in the annular space. For this purpose a large number of small holes are bored in the vessel B, and the amount of air discharged is recognized by the number of holes from which air is being liberated through the water. The outer casing C being of glass, any abnormal leakage is easily seen, and the cause then sought for.

An increase in the velocity of the water through the condenser tubes increases the rate of heat transmission between the steam and the water, and, within limits, tends to improve the vacuum. An increased velocity of flow, however, causes an increase in the power required for pumping the circulating water, and so far as the resistance of the tubes is concerned this resistance varies nearly as the velocity squared. Land practice seems to have settled down to a velocity of from 5 to 7 ft. per second for high vacua condensers, and with clean tubes and normal conditions of working the rate of heat transmission from the steam to the water may be taken to be about 600 to 700 B.Th.U. per hour, per square foot of outside tube surface, per degree Fahrenheit difference between the steam and the water. At

light loads the rate may fall to 300 or 400, and at heavy overloads increase somewhat beyond the values for normal load. The main reasons for these variations is that the air leakage usually increases somewhat as the load decreases, that is, the ratio of air to steam entering the condenser increases rapidly as the engine or turbine load decreases, and at light loads the lower tubes are more or less surrounded by a mixture of relatively stagnant air and vapour.

Condenser Calculations.—The calculation of the weight of circulating water is similar to that given on p. 221, relating to jet condensers, except that the circulating water and the water of condensation leave at different temperatures. Taking the conditions as follow: exhaust steam temperature, 105° F.; circulating water inlet, 76° F.; and outlet, 95° F.; water of condensation leaving at 100° F.; then, with the value of $L = 946$ B.Th.U. per pound,

$$W(95 - 76) = w(946 + 105 - 100),$$

$$\text{or } \frac{W}{w} = \frac{951}{19}$$

$$= 50 \text{ lb. per pound of steam condensed.}$$

With regard to the total length of tube required between the inlet and outlet of the circulating water for given conditions of operation, the following method of calculation can be used:

Let T_s = steam inlet temperature, degrees Fahrenheit.
t_1 = inlet water temperature, degrees Fahrenheit.
t_2 = outlet water temperature, degrees Fahrenheit.
K = rate of heat transmission, B.Th.U. per square foot per hour per degree Fahrenheit difference.
ρ = density of water, pounds per cubic foot.
v = velocity of water through tubes, feet per second (assumed uniform).
d_1 = outside diameter of tube, inches (steam side).
d_2 = inside diameter of tube, inches (water side).
n = number of tubes per pass (assumed the same for each pass).
l = length of tube per pass, feet.
σ = number of passes.
S = total outside tube surface, square feet.
W = circulating water, pounds per hour.

It is generally taken that the mean difference of temperature t_m between the steam and the water is given by

$$t_m = \frac{t_2 - t_1}{\log_e \dfrac{T_s - t_1}{T_s - t_2}}.$$

The heat transmitted per hour

$$= W(t_2 - t_1),$$

$$= KSt_m,$$

$$\text{or, } S = \frac{W(t_2 - t_1)}{Kt_m}.$$

Sectional area through tubes available for water flow

$$= n\frac{\pi}{4}\frac{d_2^2}{144} \text{ sq. ft.}$$

$$\therefore 3600\, vn\frac{\pi}{4}\frac{d_2^2}{144}\rho = W,$$

$$\text{or } n = \frac{4 \times 144 \times W}{3600 \times v\pi d_2^2\rho}.$$

$$\text{But, } S = (\sigma n)l\frac{\pi d_1}{12} \text{ sq. ft.,}$$

$$\text{or } l = \frac{12S}{\sigma n\pi d_1} \text{ ft.}$$

$$\text{Or, inserting for value of } n,\ l = \frac{12S}{\sigma\dfrac{4 \times 144W}{3600v\pi d_2^2\rho}\pi d_1}.$$

For example, taking the conditions given in the calculation on p. 231, where $T_s = 105^\circ$ F., $t_1 = 76^\circ$ F., $t_2 = 95^\circ$ F., and taking $w = 20{,}000$ lb. of exhaust steam per hour, $K = 600$, $v = 5$ ft. per second,

$$\text{then, } t_m = \frac{95 - 76}{\log_e\dfrac{105 - 76}{105 - 95}} = \frac{19}{1\cdot0647} = 18^\circ \text{ F.}$$

From the example on p. 231,

$$W = 50 \times 20{,}000,$$

$$= 1{,}000{,}000 \text{ lb. per hour.}$$

$$\therefore S = \frac{1{,}000{,}000 \times 19}{600 \times 18},$$

$$= 1760 \text{ sq. ft.}$$

Assuming the condenser has 3 passes, and the tubes $\frac{3}{4}$ in. outside diameter, and, say, $0\cdot65$ in. diameter inside,

$$\text{then } n = \frac{4 \times 144 \times 1{,}000{,}000}{3600 \times 5 \times \pi \times 0{\cdot}65 \times 0{\cdot}65 \times 62}$$
$$= 389 \text{ tubes per pass.}$$
$$\text{Total number of tubes} = 3 \times 389$$
$$= 1167,$$
$$\text{and } l = \frac{12 \times 1760}{1167 \times \pi \times 0{\cdot}75}$$
$$= 7{\cdot}7 \text{ ft. per pass.}$$

The condensation per square foot of tube surface per hour

$$= \frac{20{,}000}{1760}$$
$$= 11{\cdot}4 \text{ lb. nearly.}$$

Referring to steam tables, the vacuum would be $30 - 2{\cdot}24 = 27{\cdot}76$ in. of mercury when the barometer stands at 30 in.

If the outlet water temperature $t_2 = 100°$ F., with all other conditions the same, a similar calculation shows that

$$W = 792{,}000 \text{ lb. per hour,}$$
$$t_m = 13{\cdot}6° \text{ F.,}$$
$$S = 2330 \text{ sq. ft.,}$$
$$n = 308 \text{ tubes per pass,}$$
$$\text{or total tubes} = 3 \times 308 = 924.$$
$$l = 12{\cdot}9 \text{ ft. per pass,}$$
$$\left.\begin{array}{r}\text{and steam condensed per}\\ \text{square foot per hour}\end{array}\right\} = \frac{20{,}000}{2330} = 8{\cdot}6 \text{ lb. per hour.}$$

A comparison of the results in these two examples indicates how the necessary cooling surface and length of tubes increases the nearer the outlet-water temperature t_2 is made to approach the steam-inlet temperature T_s.

Any deposit of oil or dirt on the tube surfaces increases the resistance to heat transmission, and tends to reduce the vacuum. With steam turbines very little or no oil should find its way into the turbine casing, and therefore there is not much likelihood of oil being deposited on the condenser tubes by the steam. In reciprocating engines, however, oil is used in the cylinder for the lubrication of the valves and piston, and some of this oil is deposited on the tube surfaces, even though an oil separator may be used between the engine and the condenser.

The water used for circulation is sometimes very dirty, and then deposits mud or dirt on the inside of the tubes, again causing a reduction of the heat transmission. Speaking generally, however, the higher the velocity of the water through the tubes the less is this deposit likely to grow. In both cases it is necessary to clean through the condenser periodically in order to preserve a good vacuum in the condenser.

Corrosion of Tubes.—The causes of corrosion in condenser tubes are still somewhat obscure, but organic matter in the circulating water seems to facilitate the corrosion. A Committee of the Institute of Metals is at present engaged on research work on the corrosion of condenser tubes, and some interim reports have been presented. Some of the conditions which cause or facilitate corrosion are discussed in a suggestive article on "Condenser Corrosion" by W. Ramsay, F.I.C. in *Engineering*, 13th July, 1917, and in a paper read by Engineer Lieutenant-Commander G. B. Allen before the Institute of Metals, 16th September, 1920, on "Service Experience with Condensers"; see *Engineering*, 24th September, 1920.

Evaporative Condensers.—This type of condenser is serviceable mainly for small steam-power installations where there is little condensing water available at a reasonable cost. It is usually arranged as a series of pipes exposed to the atmosphere, the steam condensing inside the pipes. To facilitate the cooling action of the atmosphere, water is allowed to trickle over the outside of the pipes, and some of this water is evaporated and absorbed by the surrounding atmosphere. The amount of heat absorbed by the evaporation of each pound of water is shown by the graph in fig. 29, p. 251, and roughly it may be taken that the amount evaporated when working at normal full load is nearly equal to the amount of steam condensed.

Although the amount of cooling water required is small when compared with that needed for the other types of condensers, the evaporative condenser suffers from serious disadvantages. Large surfaces are required because of the low rate of heat transmission between the condensing steam and the atmosphere, particularly when the atmosphere has a high temperature and is already nearly saturated with water vapour. There are also numerous joints liable to leakage of air, with the result that the vacuum is often poor compared with that obtainable with the other types of condensers under normal conditions. Fans have been tried in some cases to increase the condensing capacity by increasing the velocity and amount of air passing the surfaces, but these appear to be regarded with disfavour because of the power required to drive the fans, and the cost of upkeep and renewal.

The commonest arrangement of evaporative condenser is that shown in fig. 13, as made by Messrs. Ledward & Beckett, Ltd. It is built up of a series of cast-iron pipes of corrugated section longitudinally, connected at the top to the exhaust main and at the bottom to the air-pump suction pipe. The pipes are usually 5 in. inside diameter inside the corrugations, and have a maximum diameter externally of 10 in. They are made in standard lengths bolted together at the flanges and return bends. Any convenient number of these condenser elements can be thus connected.

The condenser is usually placed in an elevated and exposed position on the top of the power-house, and sometimes it is convenient to allow the water of condensation to drain directly into the hot-well in the same manner as the water in a barometric jet condenser. A cast-iron tank is situated under the condenser to contain the body of circulating water which is raised by a centrifugal pump into the distributing trays or spreaders over the pipes.

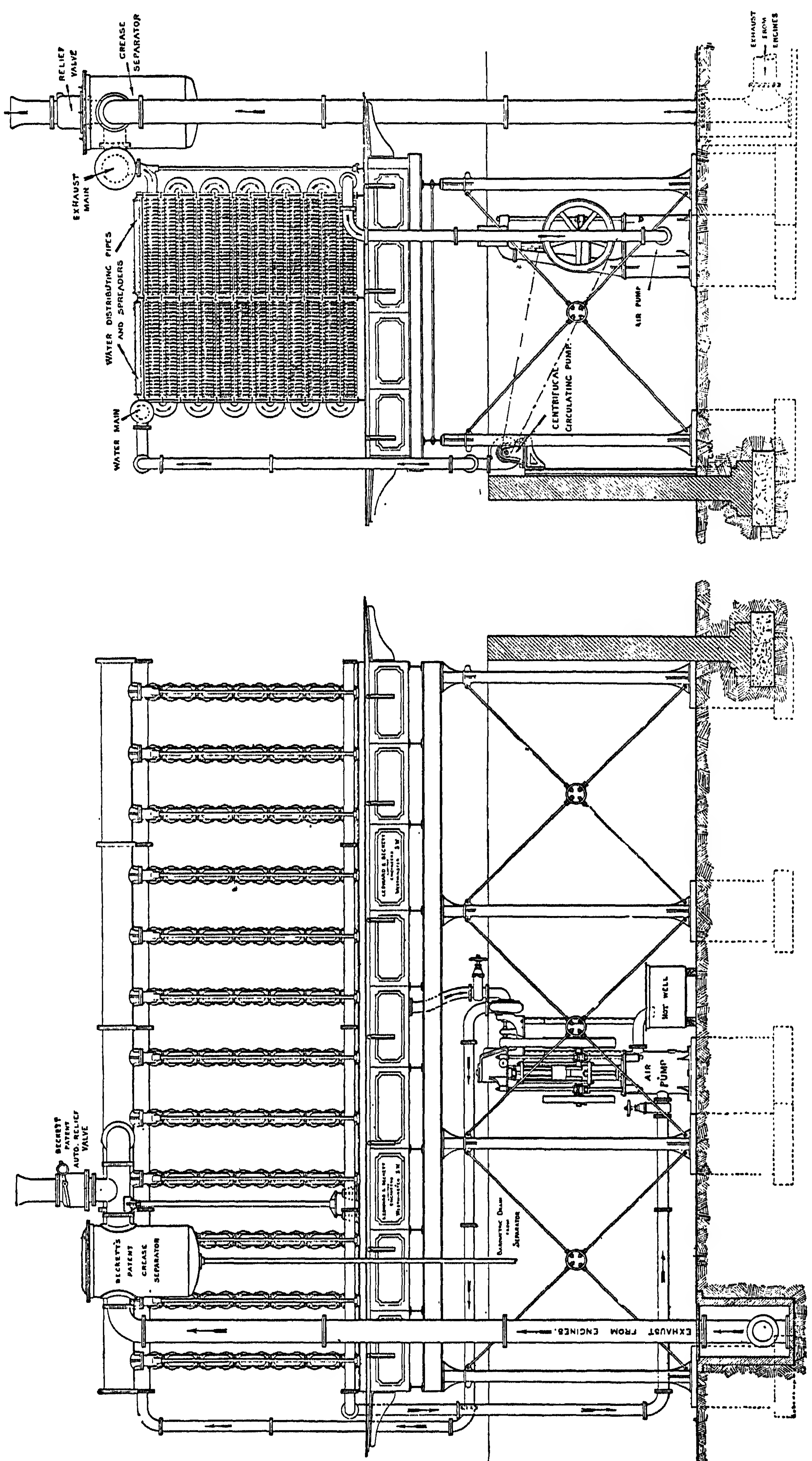


Fig. 13.—Ledward & Beckett Patent Evaporative Condenser

The amount of water circulated is usually from 10 to 20 times the amount of steam condensed, to ensure that the pipe surfaces should be kept wet from top to bottom of the condenser, and the condensation is usually from $1\frac{1}{4}$ to $1\frac{1}{2}$ lb. of steam per square foot per hour.

Comparative Merits of Jet and Surface Condensers.—The initial cost of a jet condenser is less than that of a surface condenser of the same condensing capacity, and if plenty of suitable and cheap feed water is available the jet condenser is commonly adopted, though, generally speaking, more power is required for the operation of the pumps. Sometimes the deciding factor is whether the condensing water contains salts which would form a scale on the condenser tubes, or is of a nature which would be liable to cause corrosion and pitting of these tubes, in which cases the jet condenser would have a decided advantage. But where it is desired to collect the water of condensation for boiler feed, the surface condenser would be required for this purpose. There is also the advantage in this case that the consumption of steam by the engines or turbines is readily ascertained if care is taken to keep the condenser tubes tight against leakage of circulating water.

Thus, as to whether jet condensers or surface condensers should be installed depends upon the circumstances of each case, and generally can only be decided after competitive estimates of initial cost and working costs have been made.

CHAPTER II

Air-pumps

The duty of the air-pump is to remove air from the condenser as fast as the air enters. If the same air-pump is also used to extract the water it is said to be a " wet " air-pump, but if it deals only with the air and the vapour it is termed a " dry " air-pump.

The wet air-pump is usually adopted in condensers connected to reciprocating engines, but in condensers for high vacua with steam turbines dry air-pumps deal with the air and associated vapour, and a separate pump is used for extracting the water. By the latter arrangement the water can be extracted at the highest possible temperature when required for boiler feed, while the air may be further cooled and " devaporized " either by specially arranged tubes in the surface condenser or by the injection of cold water into the air on its way to the dry air-pump. This results in a considerable reduction of the volume of the air, as is shown by the following calculations. As mentioned on p. 214, according to Dalton's law of mixtures, the total pressure in the condenser is the sum of the partial pressures of the air and the vapour. If, for example, the temperature at the air-pump suction is 86° F., and the total pressure 1·508 in. of mercury, reference to

steam tables shows that the saturation pressure at 86° F. is 1·248 in. Thus the partial pressure of the air, P_a, is

$$P_a = 1{\cdot}508 - 1{\cdot}248 = 0{\cdot}26 \text{ in.} = 18{\cdot}4 \text{ lb. per square foot,}$$

and the volume V of 1 lb. of this air is given by

$$V = 53{\cdot}2 \times \frac{(86 + 460)}{18{\cdot}4} = 1580 \text{ c. ft.}$$

If the air is saturated at 86° F., the volume of 1 lb. steam given by steam tables is 529 c. ft.

$$\therefore \frac{\text{Weight of vapour}}{\text{Weight of air}} = \frac{1580}{529} = 2{\cdot}99.$$

But if the mixture were cooled to 79° F., say, before reaching the air-pump with the same total pressure 1·508 in., the partial pressure of the air would be 0·512 in., the volume of 1 lb. air would be reduced to 792 c. ft., with a corresponding reduction of the effective displacement of the air-pump, and the weight of vapour per pound of air would be reduced to 1·21 lb.

Reciprocating Air-pumps.—A common form of air-pump is illustrated in outline in fig. 14, and is usually operated as a wet air-pump. The mixture of air and water passes from the condenser through the foot valves at the bottom on the up-stroke of the bucket. On the down-stroke a vacuum is formed on the top side of the bucket, and when the pressure there is slightly below that under the bucket the air flows through the bucket valves to the top side. Eventually the bucket reaches the water lying at the bottom of the barrel towards the end of the down-stroke, and the water also flows through the bucket valves. On the next up-stroke the air is compressed until it attains a pressure slightly greater than that over the head valves, after which it is delivered through these valves. During the final portion of the up-stroke the water also is delivered; but some water remains, filling up the clearance space.

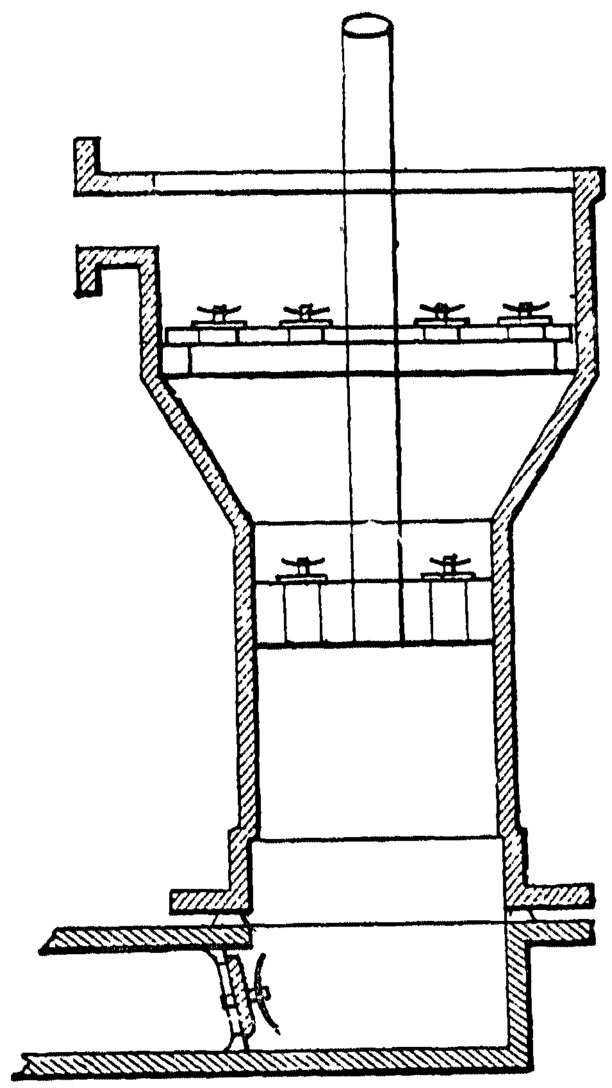


Fig. 14.—Outline of Reciprocating Air-pump

A consideration of this action shows that the foot valves are not absolutely necessary, and they are sometimes dispensed with, partly because of the difficulty of getting to these valves for inspection and repair in an emergency.

In the case of slow-speed reciprocating engines, usually an air-pump of this type is directly connected to the engine crosshead through simple

pump levers, arranged to give a bucket speed of 200 to 300 ft. per minute for an engine of moderate power, say about 1000 i.h.p. With small horizontal engines of this type the air-pump plunger is commonly operated by a tail rod from the low-pressure cylinder, the plunger then having the same piston speed as the engine. In that case the plunger is immersed in water, and arranged to act as a displacer of the water, the surface of the water then acting to draw in and compress the air.

When the condensing plant is independently operated, as is always the

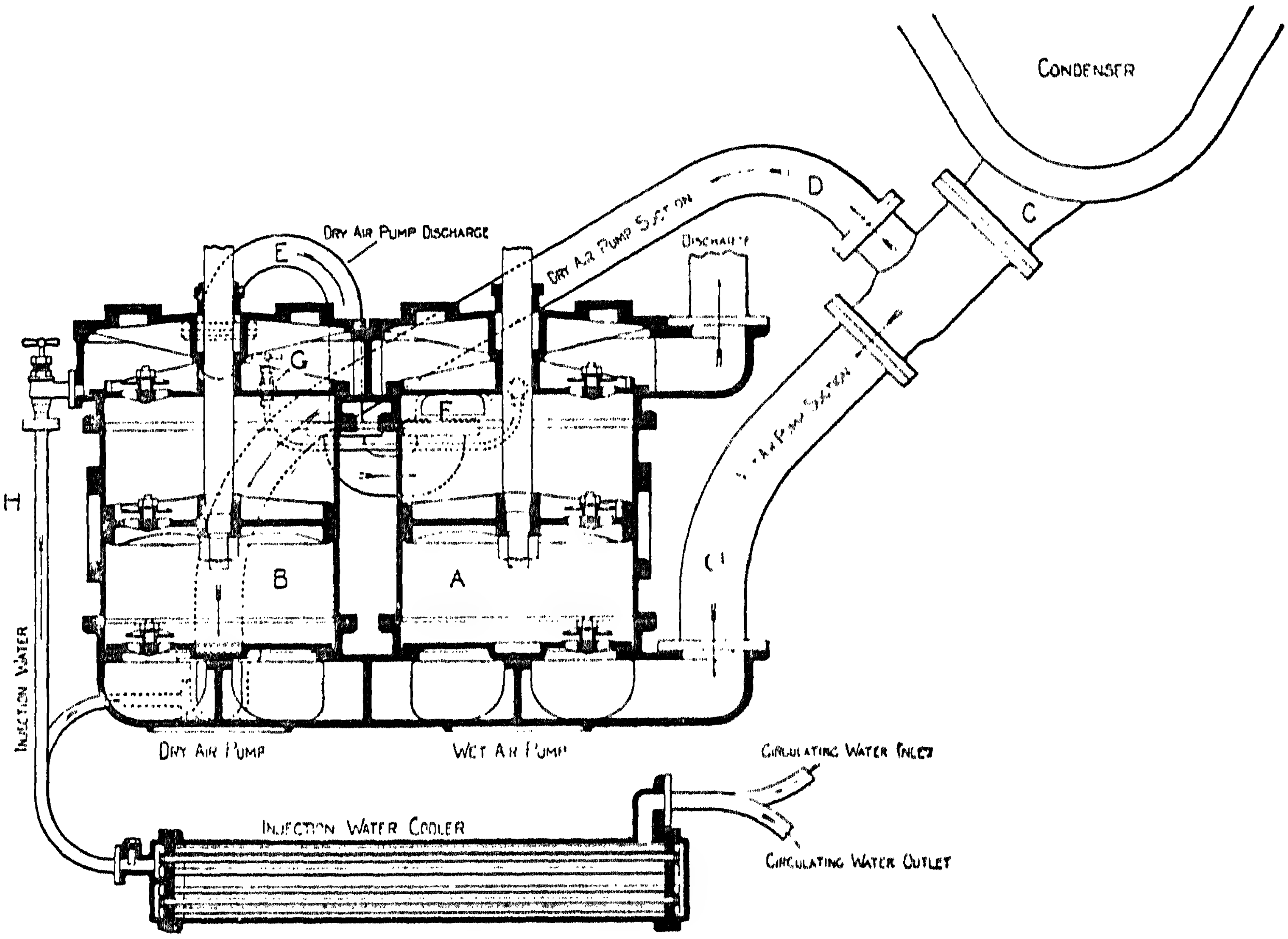


Fig. 15.—Weir "Dual" Air-pump

case with steam turbines, the air-pump may be operated either by an electric motor or by a suitable steam engine. Messrs. G. & J. Weir, Ltd. make an air-pump operated through a steam cylinder, or which, by suitable driving arrangements, can be operated by a motor. A diagrammatic arrangement of their "dual" air-pump is shown in fig. 15. If the pump is steam-driven, the steam cylinder is arranged over the wet air-pump A with its piston on the same rod, and the dry air-pump B is operated from the rod of A by levers. If motor-driven, the motor is geared to a crank-shaft, the cranks operating the pumps in the usual way.

The air-pump works in the following manner:

The water of condensation all passes by the pipe C^1 to the wet pump A, and a connection D leads to the dry air-pump B. Each pump works in the ordinary way except that the discharge from B passes along the pipe E through the spring-loaded valve F, and then into the wet pump A at a point below its

head valves, and is finally discharged by this pump. Injection water is supplied to the pump B for cooling the air and condensing the vapour, as well as for filling the pump clearances. This injection water is cooled by passing through the surface cooler shown, through which sea-water is circulated. The injection water is kept in circulation by the difference of pressure between the top and bottom of the air-pump B.

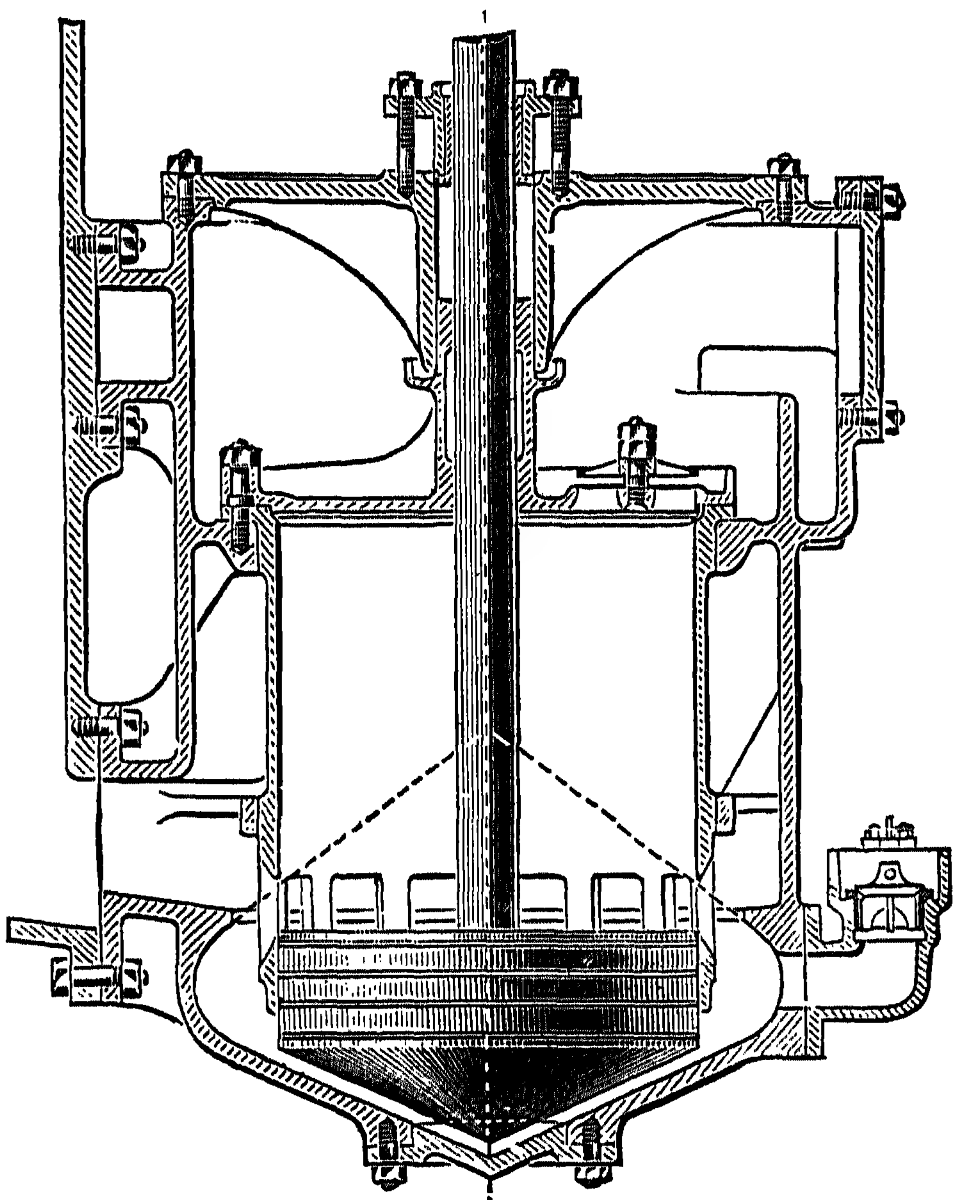


Fig. 16.—Edwards Air-pump

The Edwards Air-pump.—This type of pump is commonly run at a high speed. The bucket, as shown in fig. 16, is without valves, and as it descends the vacuum formed above the bucket is as perfect as the temperature will allow. When nearing the bottom of the stroke the conical end of the bucket strikes the water and gradually sets it in motion round the curved end of the pump barrel, the velocity acquired being sufficient to impel the water through the ports in the barrel to the top side of the bucket. Also, as soon as these ports are uncovered on the down-stroke, air from the condenser rushes into the barrel, because the vacuum there is greater than that in the condenser just before the ports are opened. Before the water has time to fall down and run back through the ports the bucket has returned and re-covered the ports, after which the air is compressed and discharged through the head valves, followed by the water. The clearance space in the barrel is practically filled with water at the top of the stroke, and on the return downstroke there is very little re-expansion.

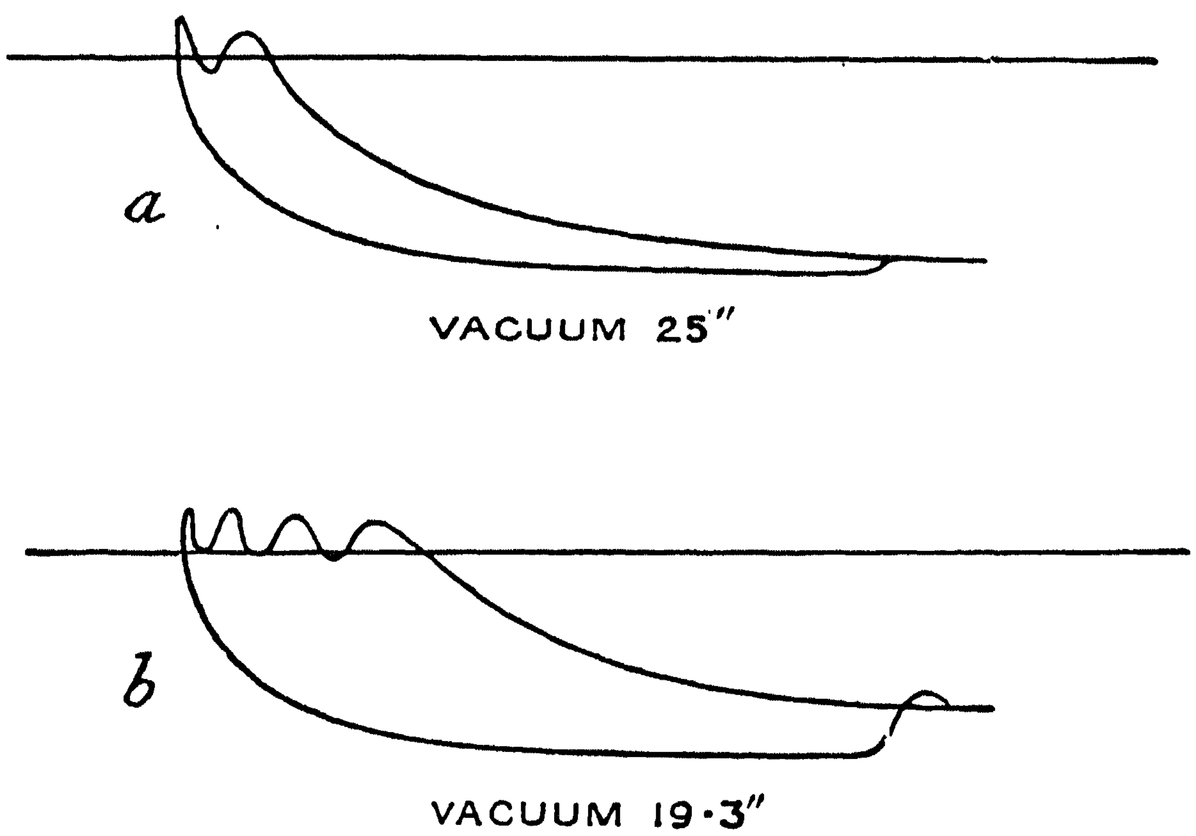


Fig. 17.—Diagrams from Edwards Air-pump

Fig. 17 illustrates the character of the indicator diagrams obtained from

the top end of a dry Edwards air-pump (*a*), when there was only a moderate leakage of air, and (*b*) when the air-leakage was abnormal. The outside views in fig. 18 refer to an Edwards triple air-pump having three cranks set at 120° and driven by a directly-connected electric motor. Usually, however, the motor drive is by a pinion and gear-wheel, so that the motor can run at a high speed. The speed of rotation of the Edwards air-pump connected to a surface condenser is commonly as high as 250 r.p.m., but if working as a wet air-pump attached to a jet condenser, the speed would

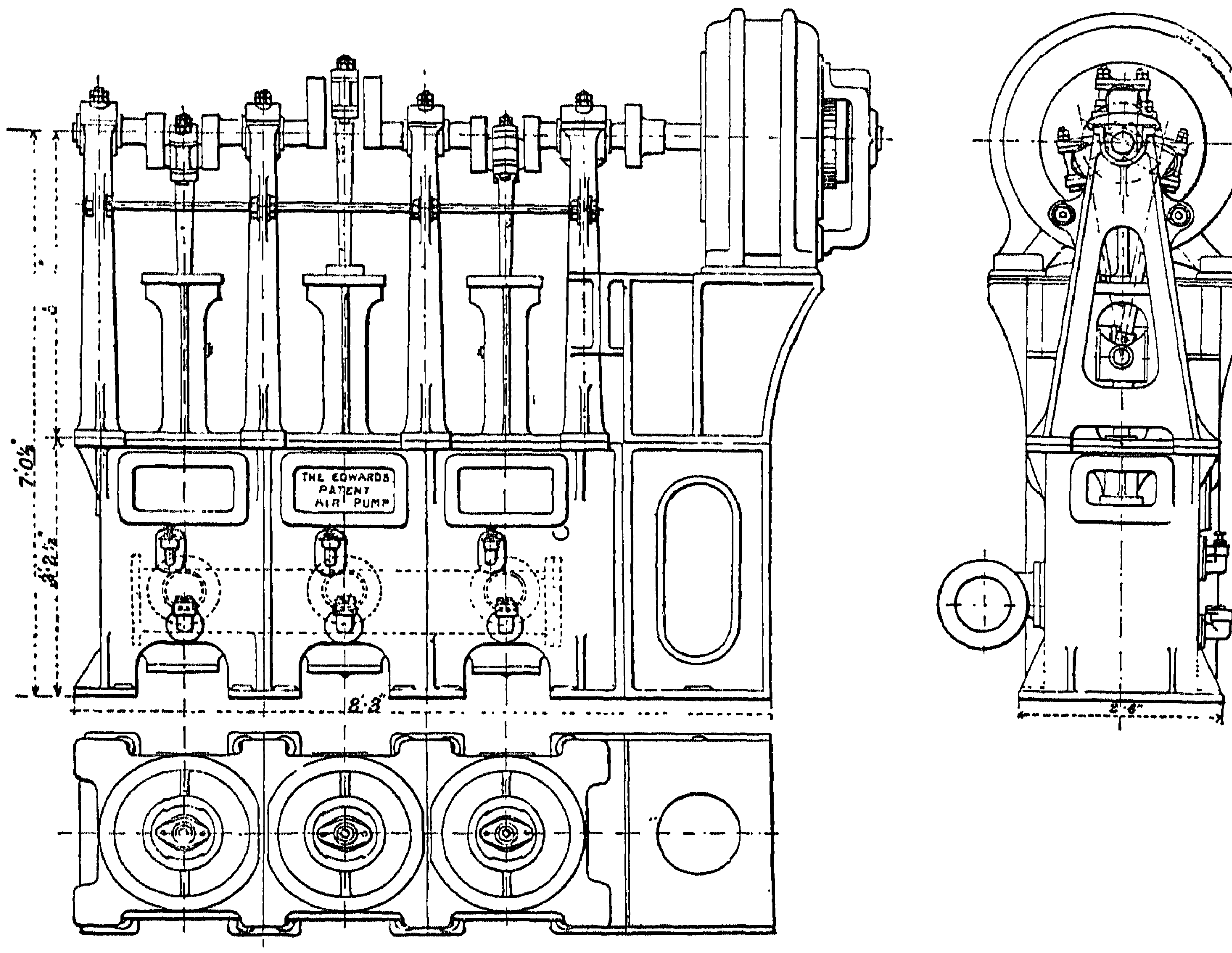


Fig. 18.—Edwards Triple Air-pump

perhaps not be more than half this value, because of the much greater volume of water to be discharged.

Fig. 19 shows a section through the cylinder of an independently-driven dry air-pump as made by The Mirrlees Watson Co., Ltd. The pump is driven either by steam or by an electric motor through a crank-shaft. The inlet of the air to the cylinder is controlled by the mechanically-operated slide-valve, and the ports are so arranged that when the piston reaches the end of its stroke, communication is made momentarily between the two ends of the cylinder, allowing an equalization of pressure. The air discharge is controlled by the voluntary opening valves arranged in the cylinder ports.

The object of equalizing the pressure at the two ends of the cylinder at the end of each stroke is to improve the volumetric efficiency of the pump. Referring to the indicator diagram in fig. 20, A L is the atmosphere

line and $a\,b$ a line at zero pressure. The compression of the air takes place along $c\,d$ and delivery along $d\,e$. The equalization of pressure now causes the pressure to fall along the line $e\,f$, and the new volume taken in on the next suction stroke is represented by the length $a\,b$. The pressure rises at c due to the air coming over from the other end of the cylinder in the manner described. Without such an arrangement the clearance volume would have

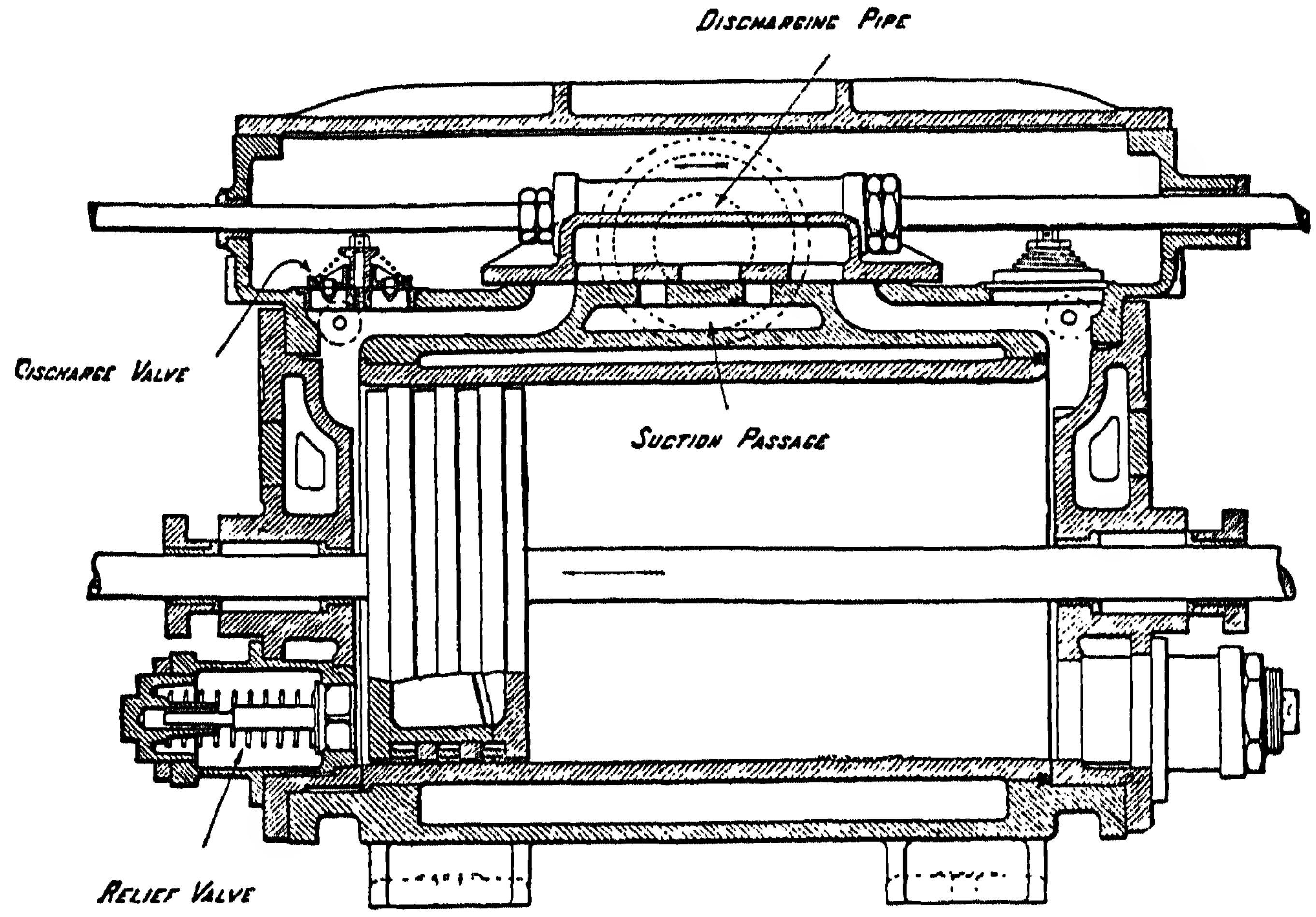


Fig. 19.—Horizontal Dry Air-pump

caused re-expansion along the dotted line $e\,h$, and the new volume drawn in would only have been represented by the length $h\,b$. It is obvious, then, that the equalization of pressure at the ends of the stroke nearly neutralizes the effect of clearance on the volumetric efficiency.

Air-pump Capacity.—The capacity of a reciprocating air-pump depends not only upon the displacement of the buckets, but also upon the volumetric efficiency. With surface condensers and steam turbines working at normal full load and under fairly air-tight conditions, the suction-stroke displacement of the air-pump may be about 0·6 c. ft. per pound of steam condensed. The higher the vacuum, however, the greater is the necessary displacement of the pump. This cannot always be obtained by increasing the speed of the pump, because the volumetric efficiency tends to fall off at high speeds. Professor Weighton's * experiments suggest that with a fairly air-tight system a suction capacity greater than 0·7 c. ft. per pound of steam condensed has very little effect on the

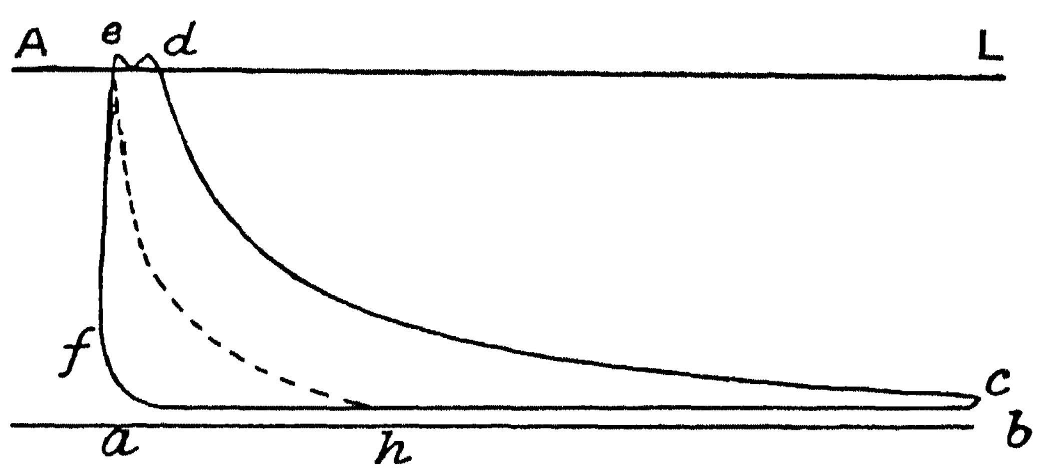


Fig. 20.—Indicator Diagram from Air-pump

* "The Efficiency of Surface Condensers", *Institution of Naval, Architects* 1906.

condenser vacuum at normal loads, but with an abnormal leakage of air, increasing the effective air-pump capacity has a decided influence on the vacuum.

In the case of jet condensers the wet air-pump requires a sufficient displacement to discharge the injection water and water of condensation as well as the air.

Rotary Air-pumps.—Many attempts have been made to produce a rotary air-pump, but probably the most successful type is the modern Leblanc air-pump. It is essentially a high-vacuum pump, and its principal characteristic is the water ejector, in which the necessary kinetic energy of the water is produced by means of a reversed turbine of partial injection. Fig. 21 shows a section through the Leblanc pump in the form now usually adopted, and operates in the following manner. Sealing-water is introduced from a tank through a suitable branch to the central chamber B, from which it passes through the water guide nozzle C. Leaving this nozzle with a comparatively low velocity, the water enters the blades D of the impeller, and is ejected into the cone E more or less in the form of thin sheets which travel with a velocity of something like 130 ft. per second. When these sheets meet the sides of the cone they form water pistons with entrapped air coming from the condenser, and the momentum is sufficient to discharge the air and water into the water-tank at a pressure slightly above that of the atmosphere, the air being liberated to the atmosphere, and the water, after cooling, allowed to return on its circuit through the pump. In the illustration the sheets of water are shown unbroken even in the discharging cone. It is hardly likely, however, that the water and air preserve these relations in the discharging cone; probably they get completely mixed up as the pressure rises towards the discharge end. The pump is conveniently driven by an electric motor or a small steam turbine directly connected.

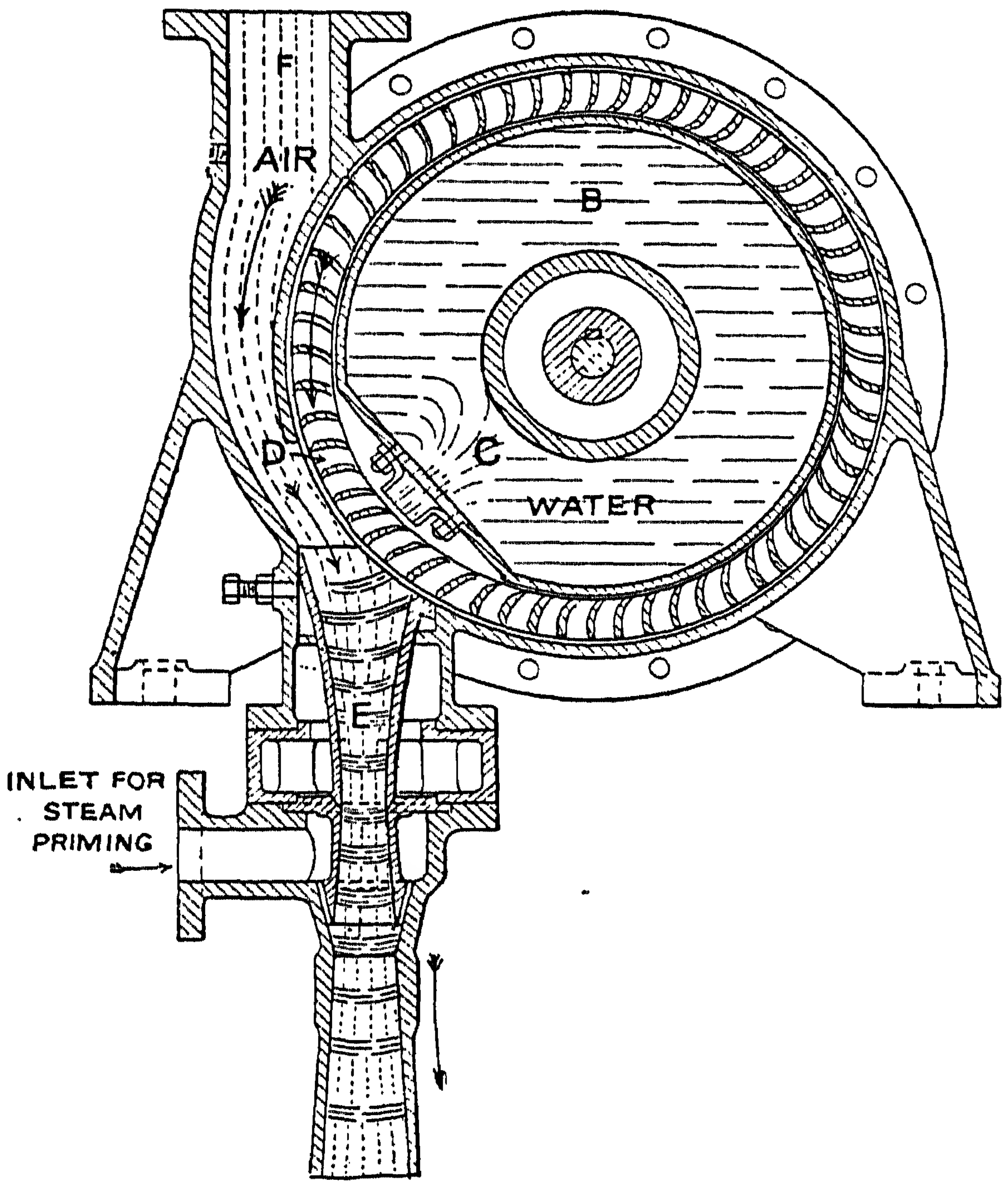


Fig. 21.—Leblanc Rotary Air-pump

A large number of these air-pumps are now at work in power-stations. Their main advantage lies in the fewness of the working parts and their

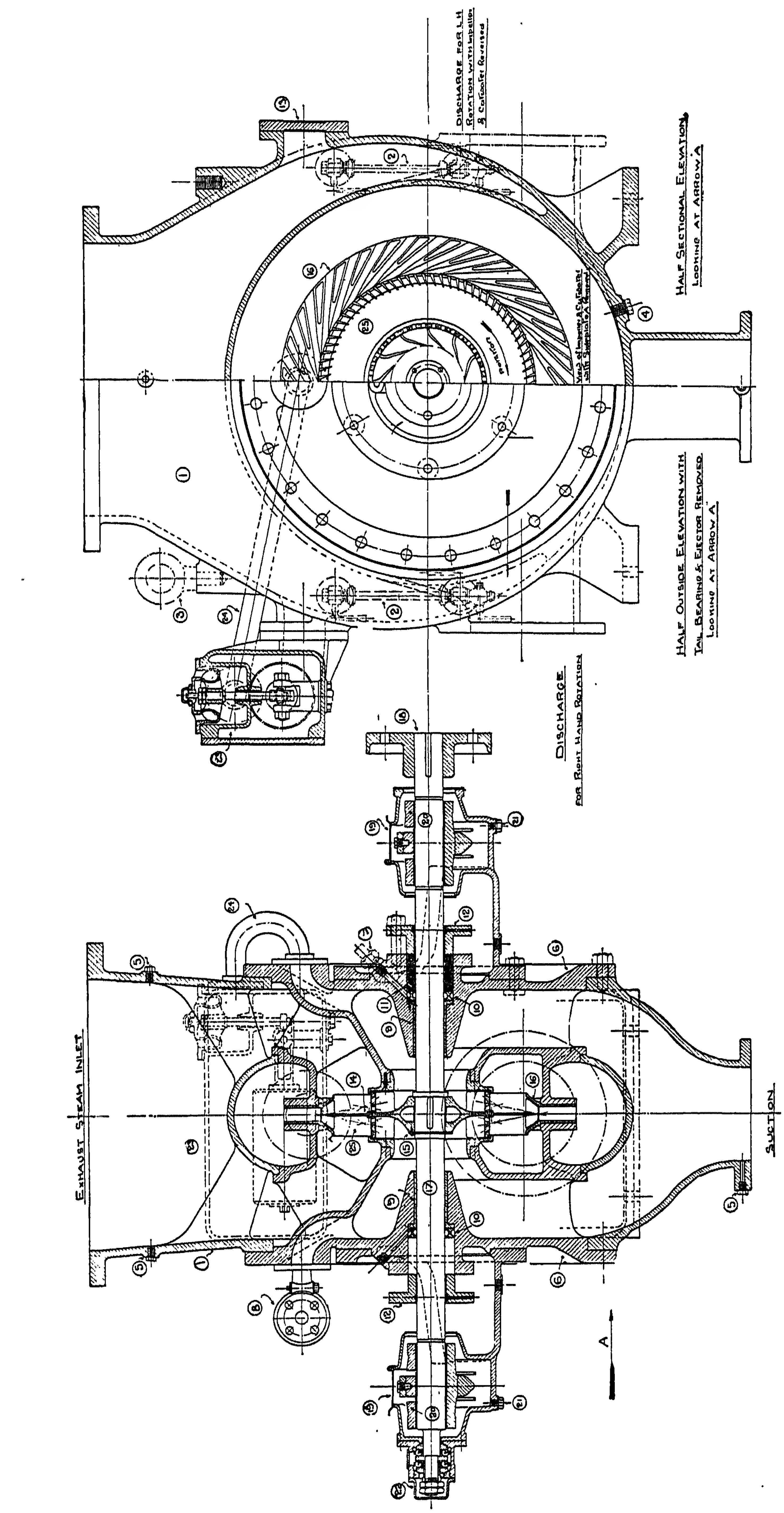


Fig. 22.—Sectional Arrangement of Rees-Roturbo Rotary Jet Condenser

freedom from breakdown. When dealing with vacua of 26 in. and lower, however, and when very large quantities of air have to be dealt with, the rotary air-pump is not as suitable or reliable as the reciprocating types previously described.

The Rees-Roturbo Manufacturing Co., Ltd. also build a rotary air-pump.

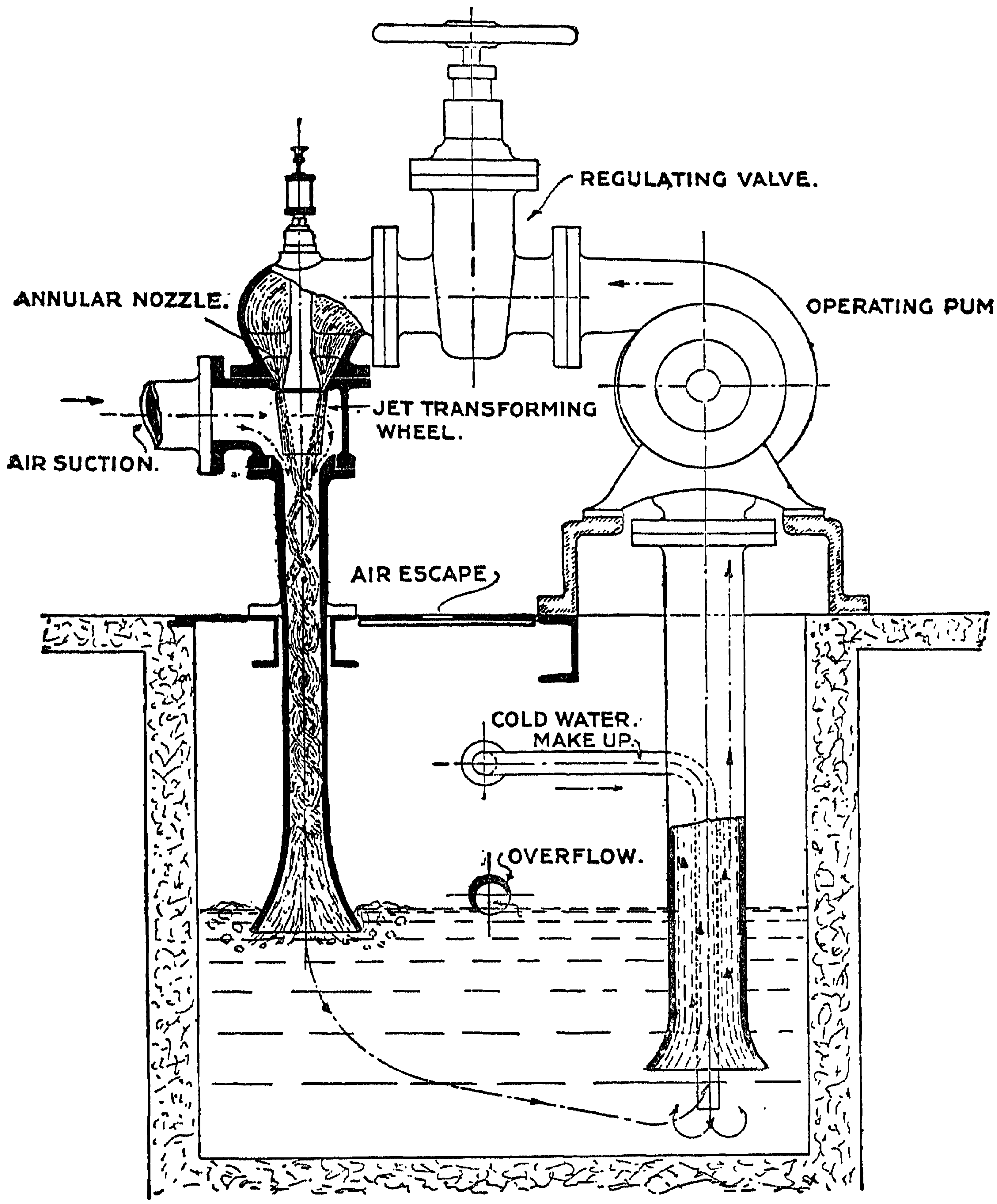


Fig. 23.—Worthington Hydraulic Air-pump

In the illustration (fig. 22) the pump is incorporated in a jet condenser, but whether used in this manner or used as a dry air-pump the action is much the same. The operating parts consist of an impeller surrounded by a set of guide blades. The central part of the impeller is a drum perforated on the circumference, and to the centre of this drum the water is supplied. The rotation of the water creates a pressure and consequently the water issues as jets from the circumferential perforations or nozzles.

The jets are arranged to impinge in pairs, and the water is thereby subdivided into fine spray and is projected across the open intervening space in the wheel to which the air from the condenser has access. In fig. 22 the exhaust steam meets with these water jets and is condensed. In any case the air is entrained by the jets of water and is carried into the circumferential blades of the rotating impeller. The velocity of the water acquired in these blades is sufficient to cause discharge through the surrounding guide blades against the external pressure.

The ejector condenser discussed on p. 219 discharges the air associated with the steam by the entraining action of the condensing water flowing at a moderate velocity through the central cone. Such an arrangement, however, is only serviceable when working as a jet condenser. Hydraulic vacuum pumps have been introduced in recent years to act solely as air-pumps. One example of this type is illustrated in fig. 23, which represents the action of the Worthington hydraulic vacuum pump. A centrifugal operating pump takes its water from a tank, and discharges the water under a suitable pressure through a regulating valve into the annular nozzle of the ejector. After leaving the nozzle the water passes through a jet-transforming wheel, by which the annular jet of water is divided up into a number of jets of approximately rectangular cross-section, leaving sufficient space between each other for the entry of air and vapour from the condenser.

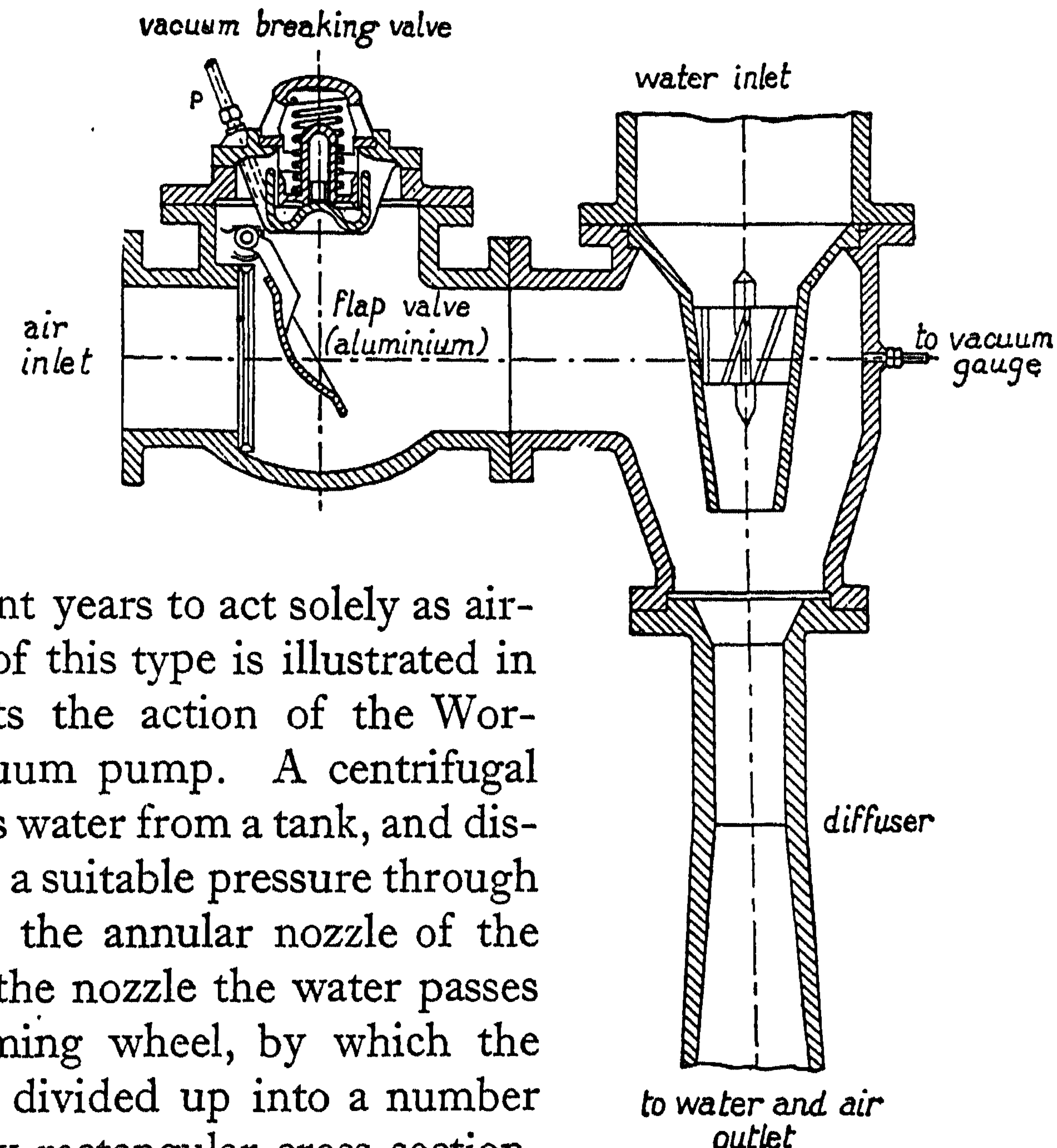


Fig. 24.—Willans-Muller Ejector Air-pump

At the same time the wheel imparts to the jets a rapidly revolving motion, as the result of which the water jets rush through the ejector cone and diffuser in the form of a helix, with the pitch and velocity diminishing as the compression of the air and vapour goes on. The jet-transforming wheel is carefully balanced and has highly-polished surfaces inside, being supported on a spindle rotating in well-lubricated ball-bearings, so that it offers practically no resistance to the flow of the water.

The water discharged by the ejector into the tank gives up the air entrained, and is circulated over again by the centrifugal pump. In order to prevent an undesirable rise of temperature a small quantity of cold water is constantly supplied to the tank, which is also provided with an overflow.

The Willans-Muller ejector air-pump operates in a similar manner, except that a separate centrifugal pump is usually dispensed with under

ordinary conditions. This ejector is built by Messrs. Willans & Robinson, Ltd. (English Electric Co., Ltd.), and fig. 24 shows it in section. The operating water is generally taken from the discharge side of the main circulating pumps, and the water in passing through the inlet cone increases in velocity, and is given a whirling motion by the fixed blades in the conical nozzle set at an angle to the axis of the nozzle. For constructional purposes these blades are attached to a central spindle, tapered at the top and bottom to reduce the resistance to the flow to the lowest possible value. The entrainment and discharge of the air is then effected in much the same manner as in the Worthington pump. Either the whole of the circulating water is allowed to pass through the ejector before entering the condenser (series system), or only a portion of the water is used in the ejector (shunt system) and is returned to the suction culvert or pipe. In any case it is found that the head of water available between the inlet and the outlet should not be less than about 16 ft., or the action is likely to be unstable.

Should the water pressure fall below that required to give proper discharge of the air and water through the diffuser there would be some danger of the water being drawn into the condenser and then into the main turbine, unless such an accidental flooding were guarded against. For this purpose an aluminium flap valve is placed between the ejector and the condenser, and is supported on a steel spindle resting on knife-edges to reduce friction. So long as the pressure at the ejector is lower than that in the condenser the valve remains fully open, but should the ejector fail and the pressure rise, the valve closes, due to the reversal of the current of air and vapour, and the communication to the condenser is thereby cut off. To prevent a large back-rush of water up the diffuser, should the valve close, which might break such a light valve by water-hammer action, a vacuum breaker is introduced in the position shown in fig. 24 in order to anticipate somewhat the action of the flap valve. A pipe P places the under side of the vacuum-breaker valve in communication with the water inlet to the ejector, and unless the pressure of the water falls unduly this valve is thereby kept closed. But should the water pressure fall to a point which would render the ejector liable to fail, as might occur if something went wrong with the pump, the spring on the top of the valve opens it and allows air to flow into the ejector from the atmosphere. If the turbine is allowed to continue running under these conditions the pressure in the condenser would rise quickly to atmospheric pressure, and then the automatic atmospheric valve connected to the turbine exhaust would open and allow the exhaust steam to flow to the atmosphere through the atmospheric exhaust main. If the pressure of the water at the ejector inlet again became normal, the vacuum breaker would close and the ejector would again begin to produce a vacuum, and then the automatic atmospheric valve would close, allowing the condenser vacuum to build up again. The vacuum breaker valve in fig. 24 is shown in the open position, but with normal running conditions would, of course, be in the closed position.

Steam Ejector Air-pumps.—With the Parsons turbine a steam jet

vacuum augmentor has been much employed, as shown in fig. 25. The condenser is inclined slightly so as to facilitate the fall of the water of condensation to the lower outlet, which is formed into a sunk trap as shown. Near the other end of the main condenser an air outlet is provided, and the air is propelled by means of a small steam ejector through a special supple-

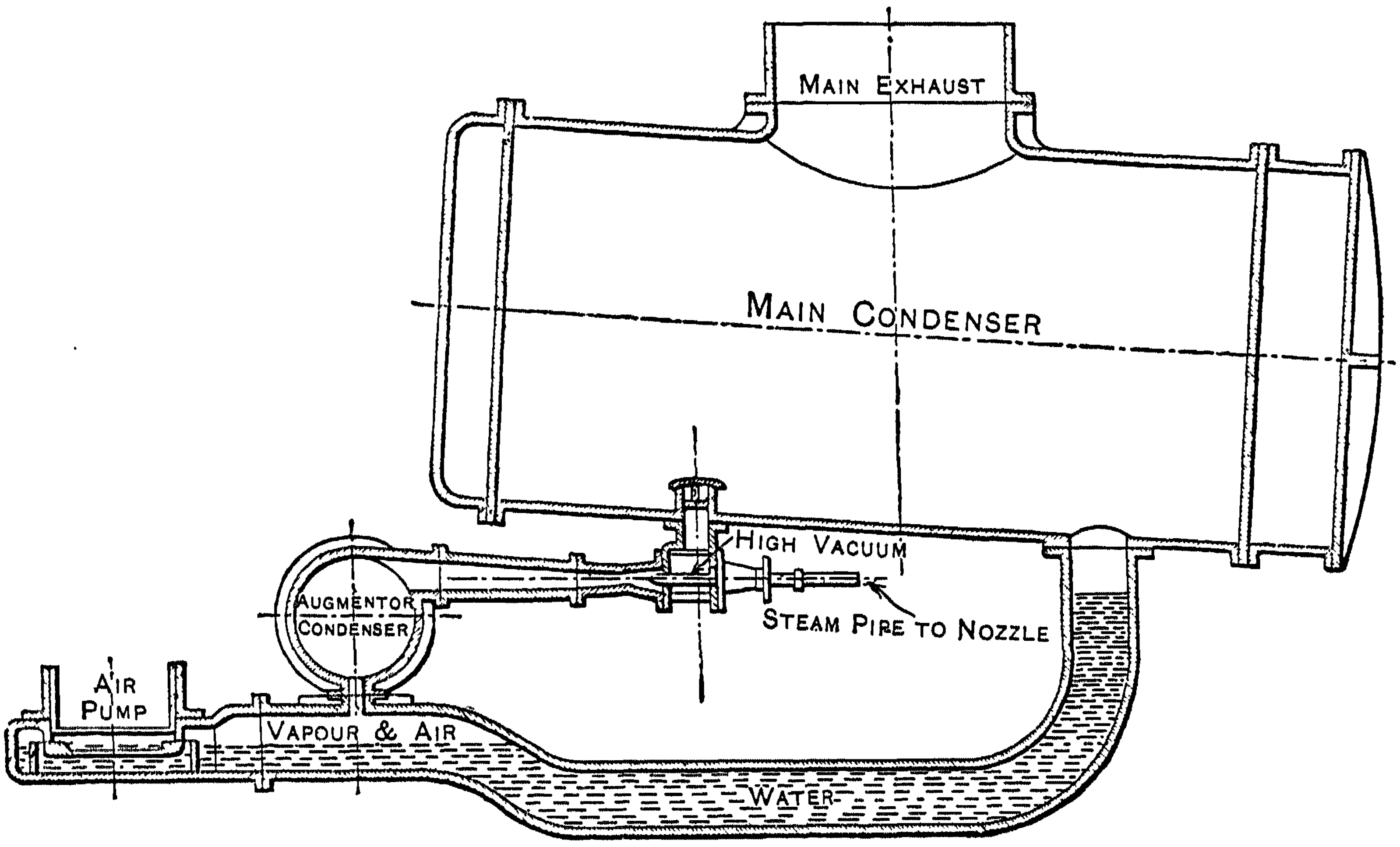


Fig. 25.—Parsons Vacuum Augmentor

mentary auxiliary condenser and delivered to the air-pump beyond the sunk water-trap. Thus the air-pump takes air already somewhat increased in density above the condenser pressure, and its efficiency and capacity is thus improved. Thus, with a condenser vacuum of $27\frac{1}{2}$ to 28 in. the vacuum at the air-pump may be only 26 in. The tube surface of the auxiliary condenser is one-twentieth that of the main condenser, and the steam consumption is said to be 0·6 per cent of the main consumption, while the vacuum is improved $\frac{3}{4}$ to 1 in., with steam turbines equivalent to perhaps 4 to 5 per cent of the main power.

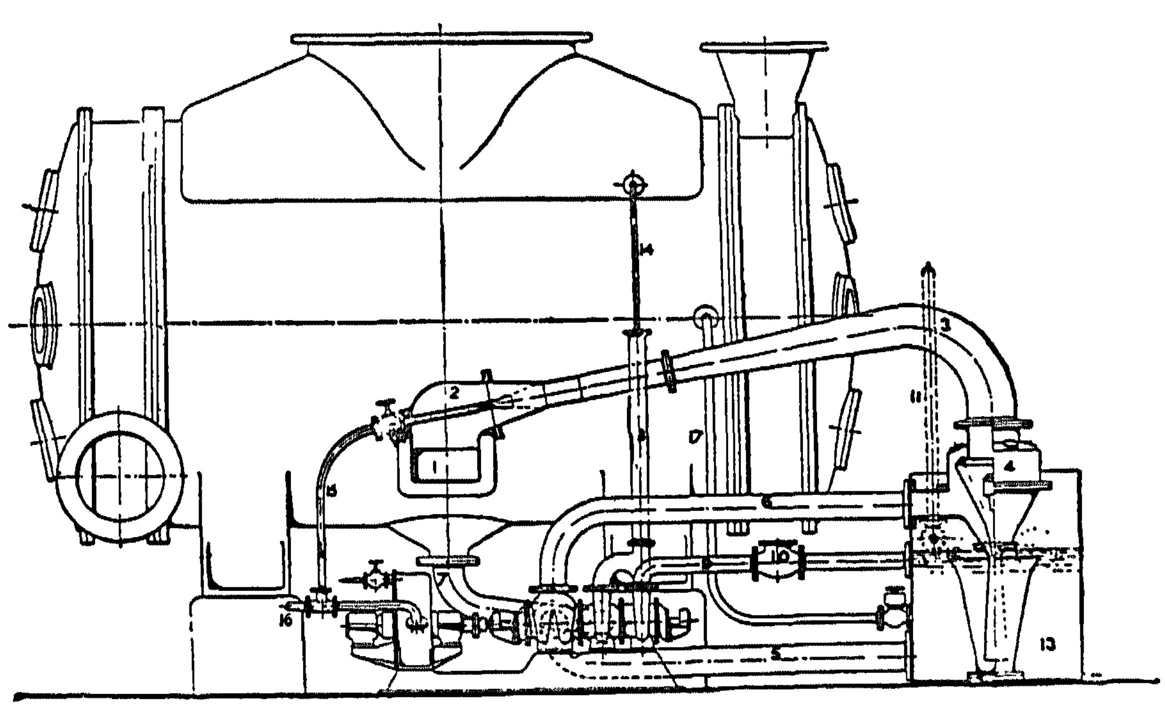


Fig. 26.—Ejector Air-pumps

In the air-pump system adopted by the Contraflo Condenser and Kinetic Air-pump Co., Ltd., a steam ejector is used to draw the air and vapour from the condenser, and to discharge it at a higher pressure into a hydraulic ejector, from which it is finally discharged to the atmosphere. The arrangement and working of the different parts of this kinetic air-pump system may be followed by reference to the sectional illustration shown in fig. 26.

In this arrangement the air is withdrawn from the condenser through the air-suction pipe 1 by means of the steam jet 2, which is supplied with exhaust steam from the driving turbine through the exhaust pipe 15 at a little above atmospheric pressure. The mixture of air and steam is delivered to the water ejector 4 through the pipe 3 at a slightly increased pressure; the steam is there condensed by the water jet, and the air is discharged into the tank 13, and through a special separator to the atmosphere. The water for the operation of the ejector is drawn from the tank by a centrifugal pump along the suction pipe 5, and is discharged along the pipe 6 to the water ejector. Here it acquires a velocity and momentum in the throat of the ejector sufficient to discharge against the pressure at the outlet (atmospheric pressure nearly), and carries the air with it.

The water of condensation is withdrawn from the condenser by a centrifugal "head" pump, and is delivered into communication with the standpipe 8. Another centrifugal pump, termed the "pressure" pump, takes this water and delivers it to the tank 13 along the pipe 9 and through the non-return valve 10. Any excess of water in the tank is discharged through the float-controlled valve 12 to the feed tanks or heater.

The steam ejector 2 may be operated by high-pressure steam if the pumps are motor-driven, the heat in any case, being returned to the boilers in the feed water, results in a rise of feed-water temperature of 5° to 8° F. at full load, and 10° to 16° F. at half-load.

The same firm have developed a combination of steam ejector and reciprocating air-pumps similar in principle to the Parsons vacuum augmentor, except that the steam used by the ejector is condensed by water of condensation from the main condenser delivered into a direct-contact auxiliary condenser from the hot well. One barrel of the independently-driven air-pump acts as a dry air-pump, taking the air from the auxiliary condenser, and the other barrel deals with the water of condensation. This arrangement is very stable in operation even with comparatively large leakages of air.

With the steam ejectors so far discussed, the ejector is only capable of compressing the air and vapour through a limited range of pressure, and the final compression to atmospheric pressure is obtained by other means. In recent years various attempts have been made to build steam ejectors capable of compressing and discharging the air against atmospheric pressure. For this purpose it is necessary to use at least two sets of steam nozzles in series. One arrangement, known as the Hick Breguet Ejectair, built by Messrs. Hick, Hargreaves, & Co., Ltd., is shown diagrammatically in fig. 27, applied to a low-level jet condenser. The air is cooled and devaporized as much as possible by the auxiliary water jet shown before leaving the condenser, and is then drawn by the primary steam jet A from the condenser, and delivered at a little higher pressure into the auxiliary condenser B. Here the steam used by the primary jet is condensed by the injection water supplied as shown. The air is then compressed by the secondary steam ejector C and delivered against atmospheric pressure. The heat in this steam may be recovered by a feed-heater. The water used in the auxiliary

condenser passes into the main condenser by the balance pipe, and is extracted along with the main injection water by the centrifugal extraction pump shown. To give stability of operation to the arrangement when the air leakage into the condenser is greater than the normal amount dealt with, an auxiliary air valve is usually fitted on the auxiliary condenser allowing a regulated amount of air to enter from the atmosphere. If an excessive leakage occurred into the main condenser, the amount taken from the atmosphere would be automatically reduced.

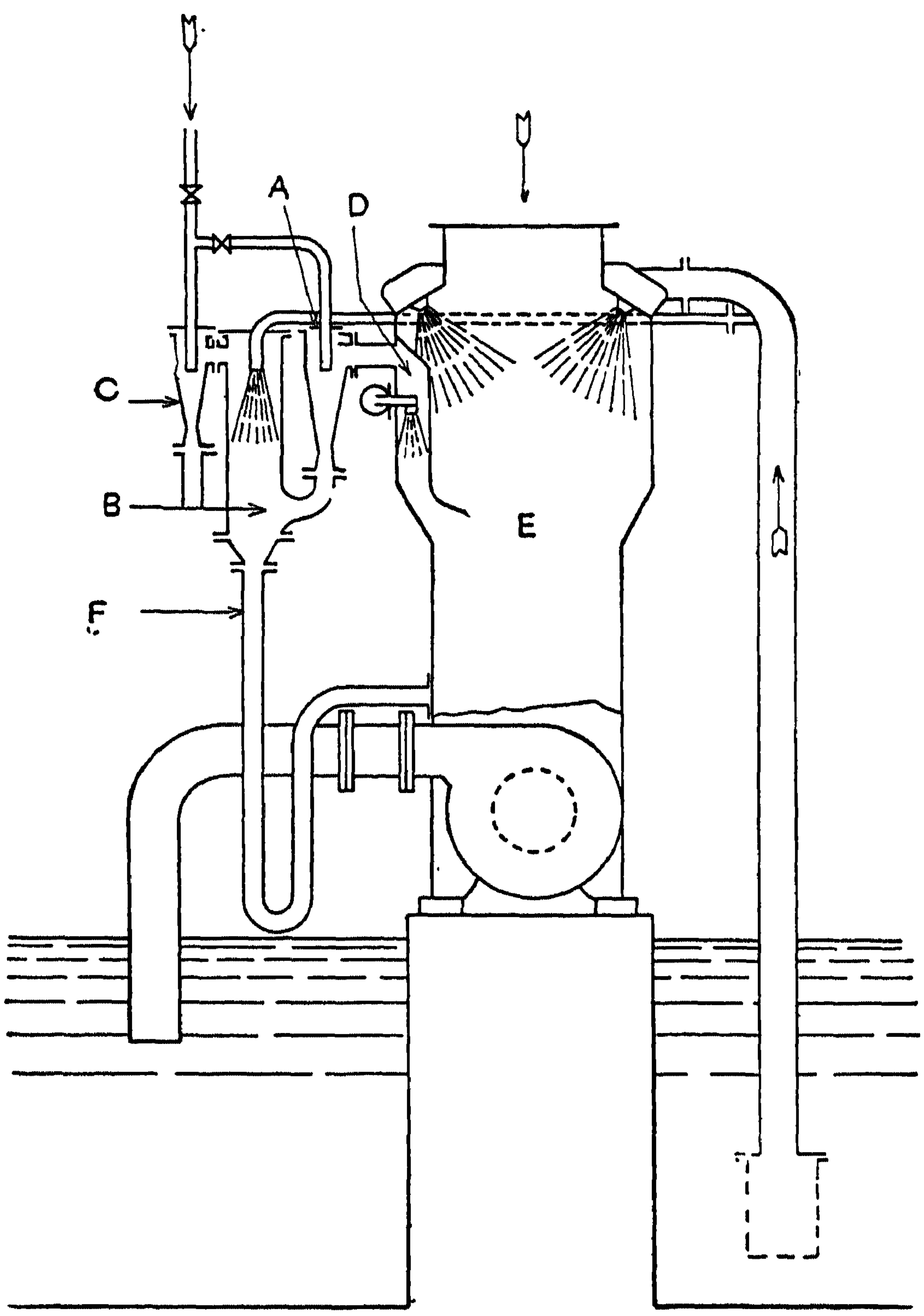


Fig. 27.—Hick-Breguet Ejectair with Jet Condenser

When the Hick-Breguet Ejectair is used on a surface condenser, water of condensation from the delivery side of the extraction pump is injected into the auxiliary condenser to condense the steam used by the primary jet A. If this is returned to the boiler as feed water, then nearly the whole of the heat in the steam used by the steam jets may be returned to the boilers.

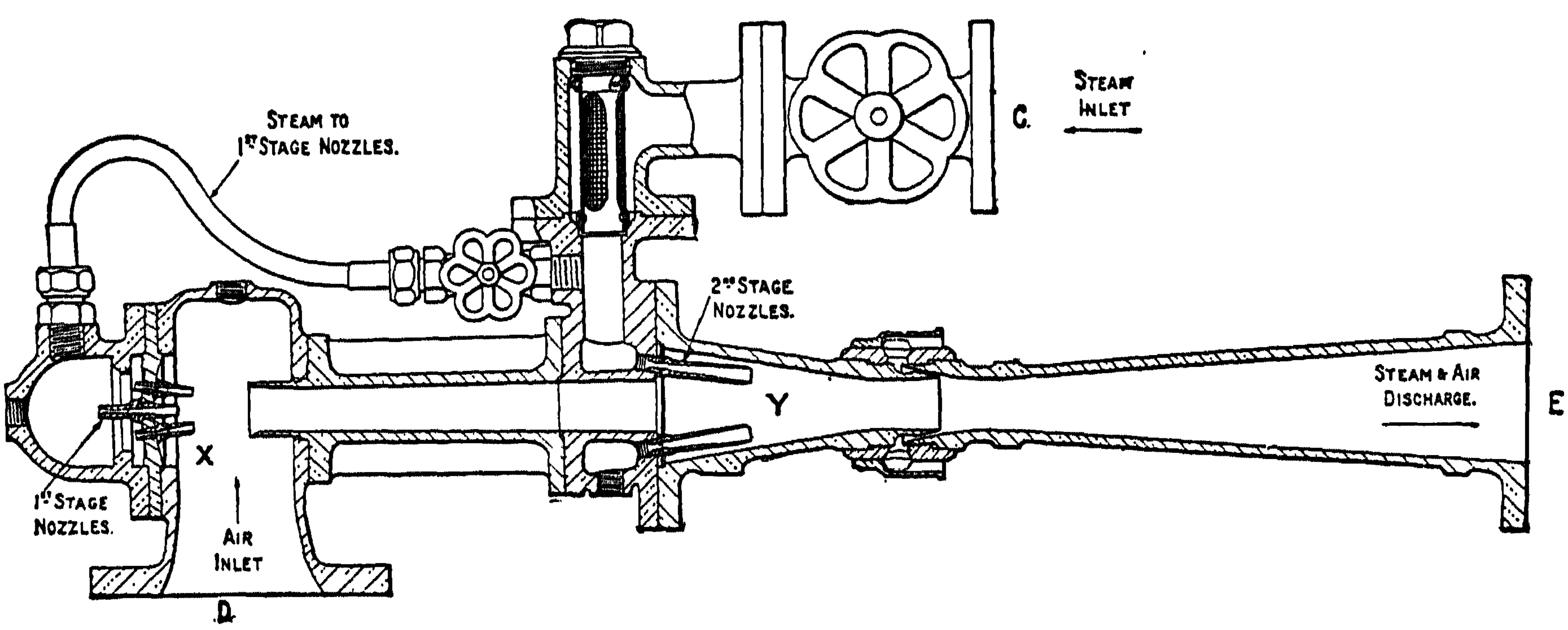


Fig. 28.—Steam Ejector Air-pump

In the Leblanc steam ejector, made by The Mirrlees Watson Co., Ltd. (fig. 28), there are two sets of steam nozzles, one, the first stage at X, and the second stage set at Y. Steam is admitted at C, and after passing

through a wire-mesh strainer enters the steam chest supplying the second-stage nozzles, and also through an auxiliary steam pipe to the steam chest of the first-stage nozzles. The air connection to the condenser is by branch D, and the steam and air are discharged at E, and may be sent into the feed-water suction tank for the recovery of most of the heat in the injector steam. The ejector is capable of discharging against a back pressure of from 14 to 16 ft. of water.

The steam nozzles are arranged in multiple form to obtain the highest possible entraining action between the flowing steam and the air. The first-stage nozzles use only a small proportion of the operating steam, about one-twentieth. The object of this set is to start in motion the air and gases, and to deliver the entire mass into the main nozzles at a high velocity. This enables the second set to increase the velocity and momentum of the stream of air to such an extent as to discharge the stream against the external pressure. The diffuser, being designed to deal with a greater amount of air than normally leaks into the main condenser, does not work well when dealing only with small amounts of air. To ensure greater stability of operation, some leakage of air from the atmosphere into the throat of the diffuser is allowed. This leakage reduces automatically should there be an abnormal amount of air coming from the main condenser.

CHAPTER III

Water Cooling and Cooling Towers

Water Cooling.—The calculations on pp. 221 and 231 will have shown what a large amount of water is necessary for condensing purposes, particularly when high vacua are required. The availability of cooling or condensing water often determines the site of the power house or station, for if it can be placed where there is little or no danger of failure of the water-supply, the problem of providing the water is greatly simplified. Frequently there is no natural or cheap supply of water available in quantity at all seasons, and some system of cooling must be adopted. There are three methods in common use for this purpose, viz. the pond or reservoir, the sprayer, and the tower. These arrangements will be considered later in detail.

Whatever the system of cooling adopted, the principal cooling action depended upon is the absorption of vapour, and the equivalent latent heat, from the surfaces of the water by the atmosphere or air in contact with or near to these surfaces. The heat thereby taken up is obtained at the expense of heat in the water remaining, which cools in consequence. The amount of vapour contained in air saturated at any particular temperature may be calculated in the manner given on p. 237, and the corresponding amount of heat is easily estimated. The results of such calculations per pound of air at atmospheric pressure are shown in fig. 29, and it would be noted how

rapidly the vapour contents and the corresponding amount of heat increase with the temperature.

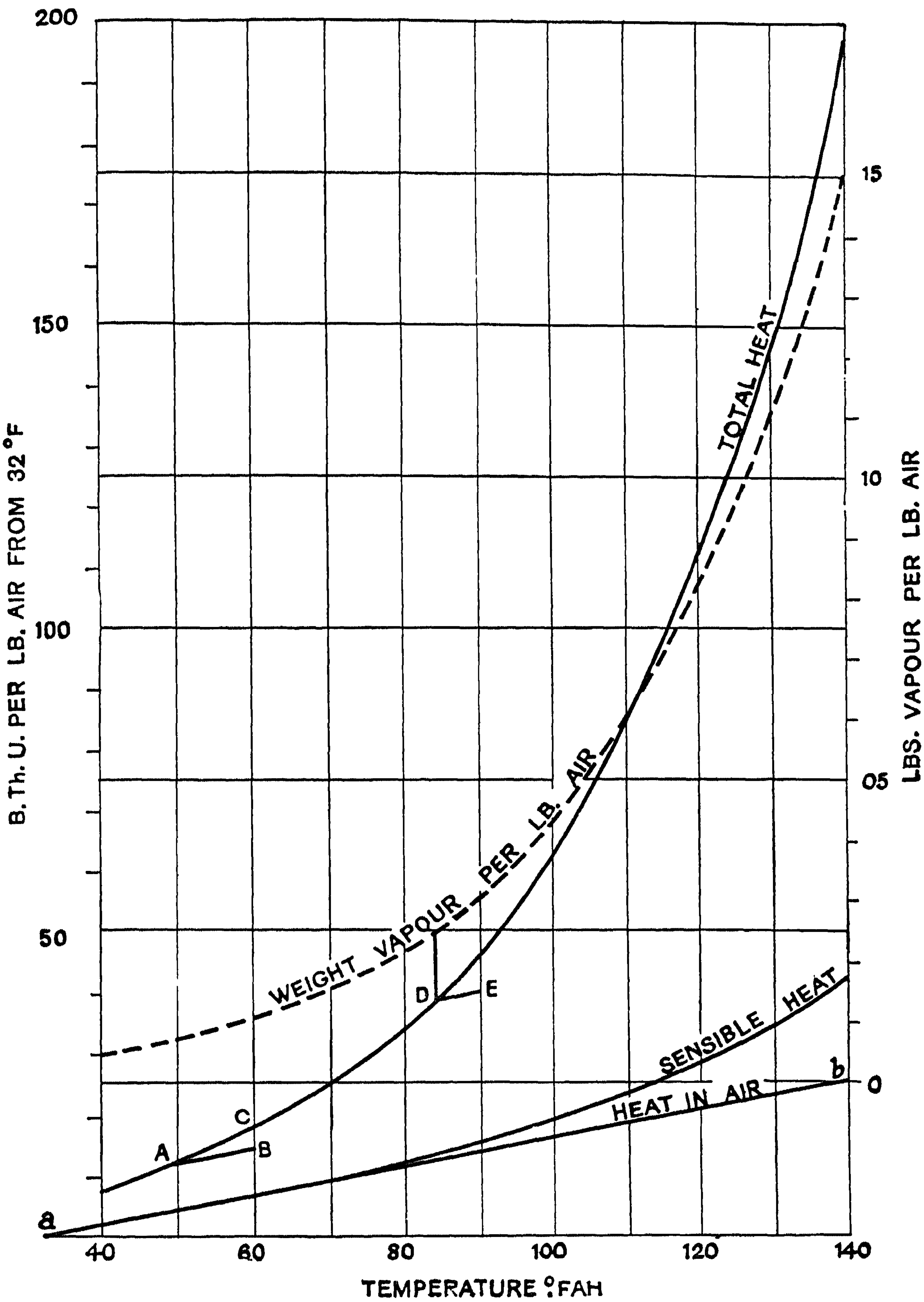


Fig. 29.—Heat and Vapour Contents in 1 lb. Air at Atmospheric Pressure

Suppose the saturated air entering a cooling tower at 60° F. leaves the tower saturated at 90° F., then the amount of vapour absorbed is

$$0{\cdot}031 - 0{\cdot}011 = 0{\cdot}02 \text{ lb. per pound air,}$$

and the absorption of heat is

$$48 - 18 = 30 \text{ B.Th.U. per pound air.}$$

Under normal conditions of operation, however, the air entering and

leaving is rarely saturated, but its humidity may be estimated by means of a hygrometer, the one usually adopted for this purpose being the wet and dry bulb thermometer type. If the air entering at 60° F. had a dew-point temperature 50° F., say, the vapour contents would be 0·008 lb. per pound air. To find the heat contents the line A B (fig. 29) would be drawn parallel to the straight line *a b*, giving the point B, the heat contents being 15 B.Th.U. The small amount of heat represented by the superheating of the vapour from 50° F. to 60° F. is neglected in this estimate. If the air at outlet had the relative humidity 80 per cent, and the temperature is 90° F., the vapour contents at outlet would be

$$0{\cdot}031 \times 0{\cdot}8 = 0{\cdot}0248 \text{ lb. per pound air.}$$

This is a dew-point temperature of 84° F., and the heat contents is again given by drawing the line D E parallel to *a b*, giving 40 B.Th.U. per pound air. Therefore the absorption of vapour is

$$0{\cdot}0248 - 0{\cdot}008 = 0{\cdot}0168 \text{ lb. per pound air,}$$

and the heat absorbed is

$$40 - 15 = 25 \text{ B.Th.U. per pound air.}$$

If the heat given up at the condenser is 950 B.Th.U. per pound steam condensed, and the water under steady conditions is cooled from 100° F. to 75° F., then the water per pound steam condensed is

$$\frac{950}{100 - 75} = 38 \text{ lb.}$$

The amount of water cooled per pound air is

$$\frac{25}{100 - 75} = 1 \text{ lb.,}$$

and therefore the amount of air required per pound of steam condensed is given by

$$\frac{38}{1} = 38 \text{ lb.}$$

The evaporation of water is

$$\frac{0{\cdot}0168 \times 100}{(1 + 0{\cdot}0168)} = 1{\cdot}65 \text{ per cent,}$$

and make-up supply equivalent to this would be necessary with surface condensers, or with jet condensers the fresh feed water supplied to the boilers would probably be sufficient for continuous operation under these conditions.

Similar calculations to those given above show how much more difficult it is to cool the water for high-vacuum conditions than it is for low vacua. For high vacua, because of the relatively low temperature of the water

leaving the condenser, the humidity of the entering air at the cooler becomes an important factor, whereas when low vacua only are required, with the consequent higher water temperatures at condenser outlet, the humidity and temperature of the entering air is not so important.

Cooling Ponds or Reservoirs.—Where there is sufficient ground area available for a cooling pond, without excessive cost, this arrangement has been commonly adopted for mills and works when no natural supply of condensing water is available. The construction of such a pond, however, is somewhat expensive in first cost and it is now a common practice to limit the size of the pond, and to assist the cooling of the water by introducing one or other of the other methods of cooling as an auxiliary. In the usual arrangement a low-level jet condenser would preferably be adopted, with the pond at such a level that the water will flow freely into it from the hot well. As mentioned on p. 222, the vacuum in the condenser may be depended upon to lift the water through a moderate height and inject it into the condenser.

Although a cooling pond is costly to construct, it requires little attention, and costs little for upkeep unless it should become leaky. To prevent leakage of water the sides and bottom usually have to be puddled with clay for a thickness of 18 to 24 in., and the sloping sides finished off with rubble. The coursing may be thickened near the water level and set in hydraulic cement for a depth of a foot or so, to prevent rats from boring holes into the embankment. The depth of the pond would depend to some extent upon the conditions of working and upon the amount of make-up supply likely to be available during a drought, but in any case it is hardly desirable to have the pond deeper than the lowest adjacent drain, so that it may be completely emptied for cleaning out of the accumulations of mud and dirt when required. Speaking generally, if the engine runs only during the daytime a deep pond of relatively small cooling surface would suffice, whereas a night and day load requires a pond of larger cooling surface, but it may be comparatively shallow.

The pond should be clear from buildings and trees, so that the wind may have free access over the surface, as this increases the cooling action considerably, and any fencing should be as open as possible and placed several feet from the edges for the same reason. The hot water is usually carried to the farthest end of the pond in a shallow trough and taken to the condenser from near the bottom at the other end through a perforated pipe, commonly called a " snore " pipe, thus preventing leaves, weeds, &c., from being carried into the condenser and air-pumps.

When the engine works only on a day load, the area of the pond may be made about 33 sq. ft., and the capacity about 200 c. ft. per indicated horse-power. Details of construction may be seen in a paper on cooling ponds by Mr. H. W. Barker.*

Spray Cooling.—The main object in spraying the hot water into the atmosphere is to expose a large surface to the cooling action of the air. The

* *Proceedings Institution of Civil Engineers*, Vol. CXXXII.

water is pumped through spraying nozzles screwed into a system of pipes placed over the pond. A head of about 20 ft. of water is required for this purpose, and this represents the net expenditure of 20 ft.-lb. of work on each pound of water pumped. In some arrangements the spraying nozzles eject the water upwards and sideways, and it then falls into the pond or tank as a fine shower. In other arrangements the nozzles are in an elevated position and spray downwards, but this generally absorbs extra power in pumping, and may require an enclosure of boards arranged louvre fashion to prevent some of the spray being carried away by the wind in crowded districts. One form of spraying nozzle is shown in fig. 30, as made by Messrs. Ledward & Beckett, Ltd. The issuing water is given a rotary motion, causing it to spread out on leaving the nozzle, and it therefore splits up readily into small drops.

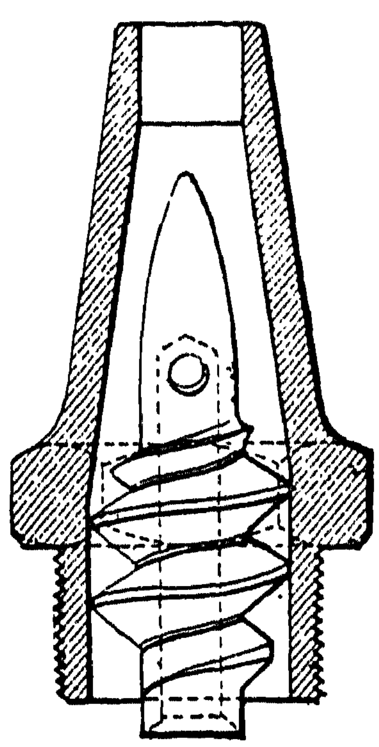


Fig. 30.—Water Sprayer

The superficial ground area required for efficient spray cooling may be taken at about 1 sq. ft. per 5 lb. of steam condensed per hour with a vacuum of about 26 in. If necessary this area may be reduced, but at the expense of reserve cooling area in calm weather.

Cooling Towers.—Where sufficient land for a cooling pond is not available, or is too dear, the cooling of the water for a large power plant is sometimes a difficult problem. For a small plant a simple cooler may be adopted consisting of thin boards set louvre fashion and exposed to the air and winds, and over these the water is allowed to trickle. The water is distributed by a trough at the top of the cooler and eventually falls into the tank or pond below. Two sets of boards may be used, placed at right angles to one another, so that, whatever the direction of the wind, one set or other will be fully exposed to it, or both sets will be partially so, the wind passing through the spaces left by the louvre formation of the boards. In a crowded district it may be necessary to surround the cooler with a louvre frame of wood to prevent the wind carrying water spray away.

Chimney coolers are usually adopted for large powers. The water is pumped to a height of about 25 ft. and descends by gravity, being distributed by special troughs to fall on to splashbars, thin boards, or drain tiles. Above this is arranged a wooden chimney or tower, as shown in fig. 31. The draught of air is created by the chimney effect of the heated air and vapour inside the chimney, and this action is the more intense the hotter the water, being so far more or less self-regulating. The chimney is usually carried to a height of 60 or 70 ft.

The arrangement adopted by The Premier Cooler and Engineering Co., Ltd., is shown in fig. 31. The condensing water is delivered to a central trough, and is then distributed by auxiliary troughs running at right angles to the main one. A series of nozzles in the bottom of these troughs distributes the water on to the top of splashing plates set directly under-

neath, as shown by the detail illustration in fig. 32. From these the water rebounds and falls on to the triangular splashbars below. Some of the water rebounds from each splashbar and some runs down to the lower edge

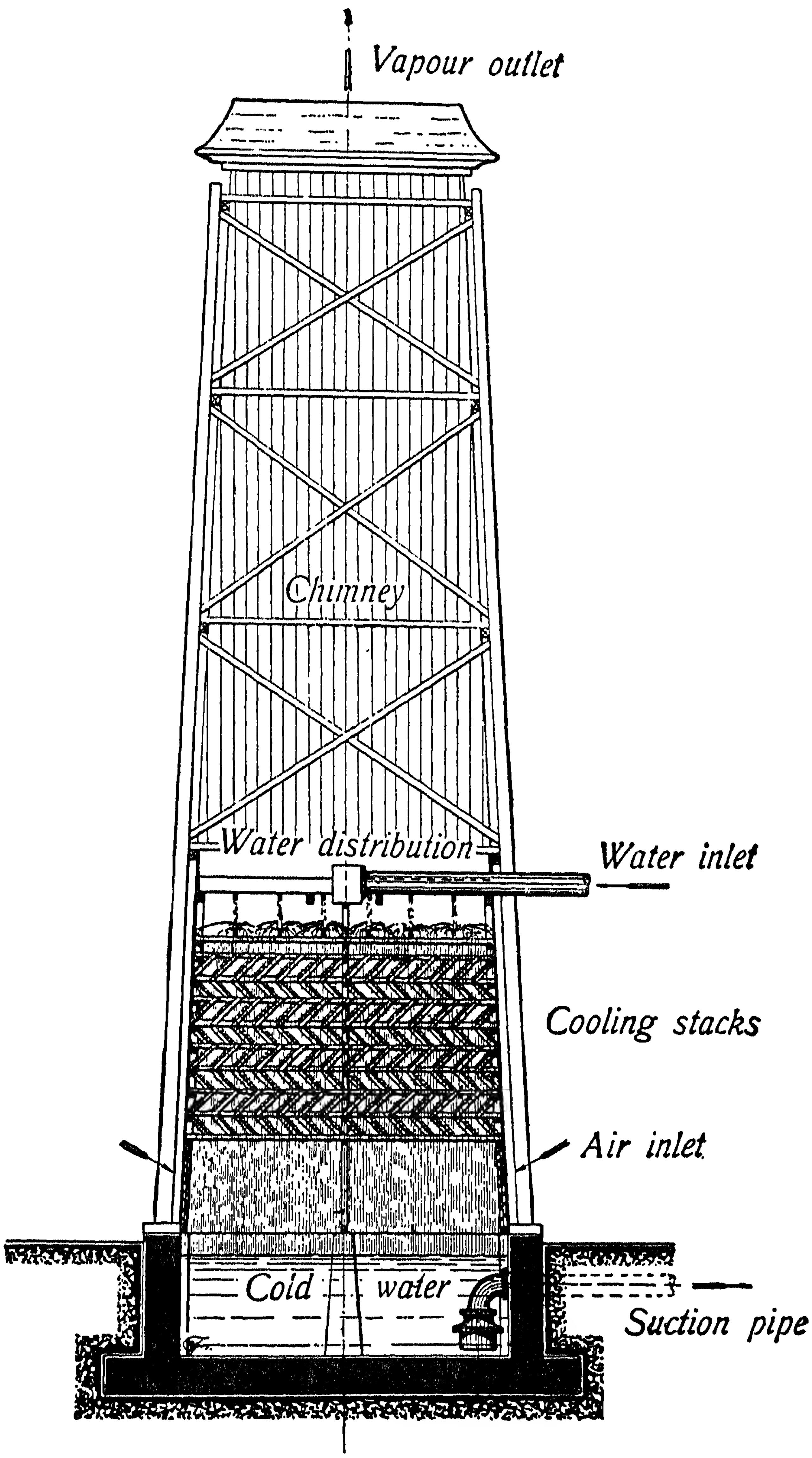


Fig. 31.—Chimney Cooler

of the bar and falls on the bar below. Eventually the water falls into the cooler tank below as a fine shower.

Several other firms build similar cooling towers, the principal differences being in the arrangement and shape of the splashbars. The Davenport

Engineering Co., for instance, use diamond-shaped splashbars supported in hurdles and grouped as shown in fig. 33. Messrs. Richardson, Westgarth, & Co., Ltd., adopted splashbars with convex top after the results of experiments by Mr. I. V. Robinson.* The method of experiment adopted consisted in allowing 50 drops of coloured water to fall in one minute from a height of 36 in. on to the length of bar under test. The water which rebounded was allowed to fall on a sheet of white paper. Counting the spots on the paper and noting their general positions led to the adoption of the section above-mentioned. The under side of each bar has cross-cuts

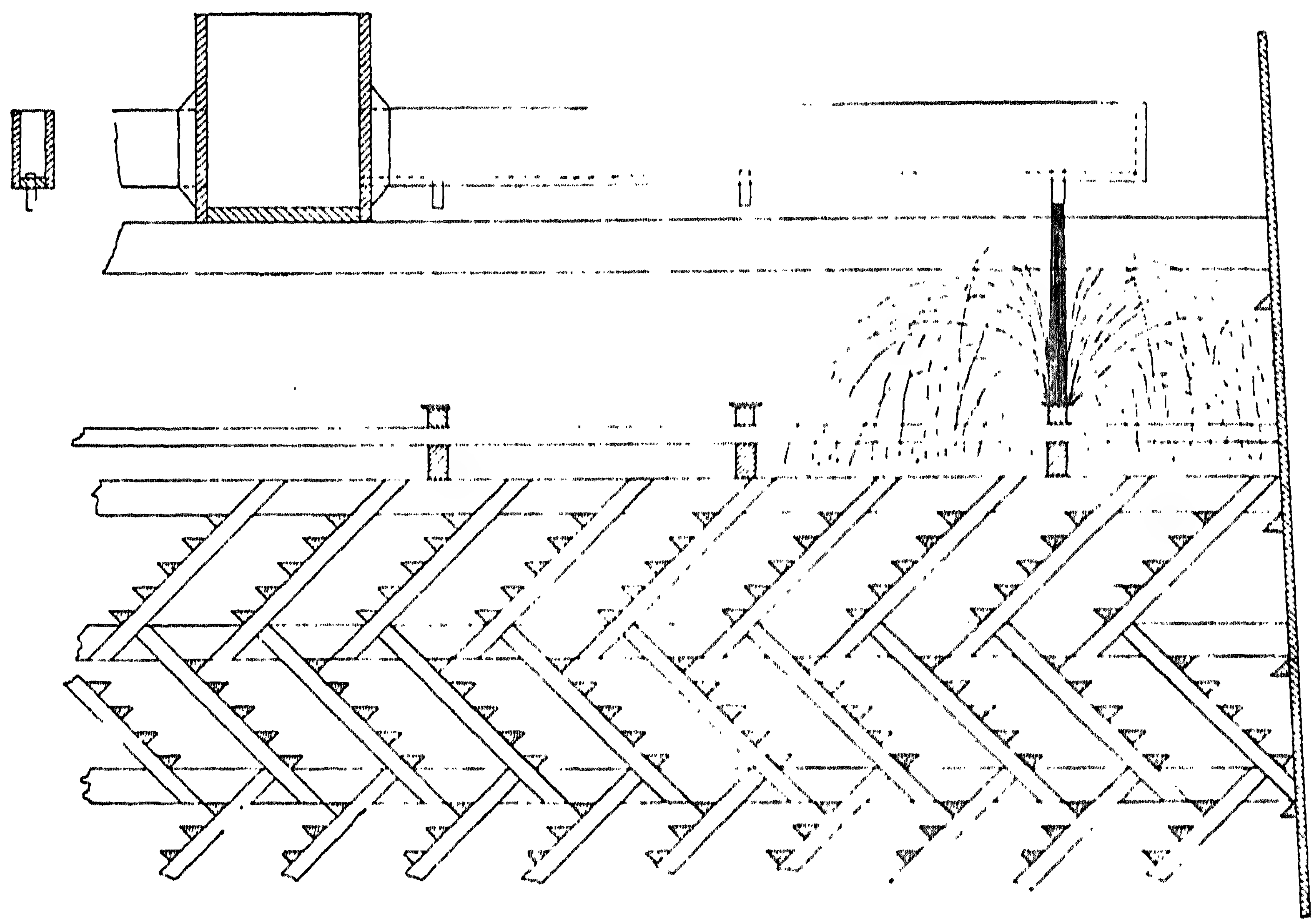


Fig. 32.—Arrangement of Splash-bars, &c.—Premier Cooler and Engineering Company

spaced about $1\frac{1}{2}$ in. apart to ensure that the water would drop off in fine streams.

All the timber forming the structure of the tower should be treated with some preservative such as creosote or sideroleum. The boarding is placed on the inside of the framework, so as to give as smooth a surface as possible inside, and these boards should fit close together to prevent ingress of cold air in the chimney. All the bolts and plates used are preferably galvanized to save frequent painting. The main framework needs to be well anchored down to stable foundations to withstand the overturning force or moment due to wind pressure. For calculation purposes the wind pressure may be taken at about 60 lb. per square foot of section exposed to the wind.

The size of a chimney cooler depends largely upon the amount of water to be cooled, the fall of temperature required, the amount of air which can

* "Cooling Towers", West of Scotland Iron and Steel Institute, 1907.

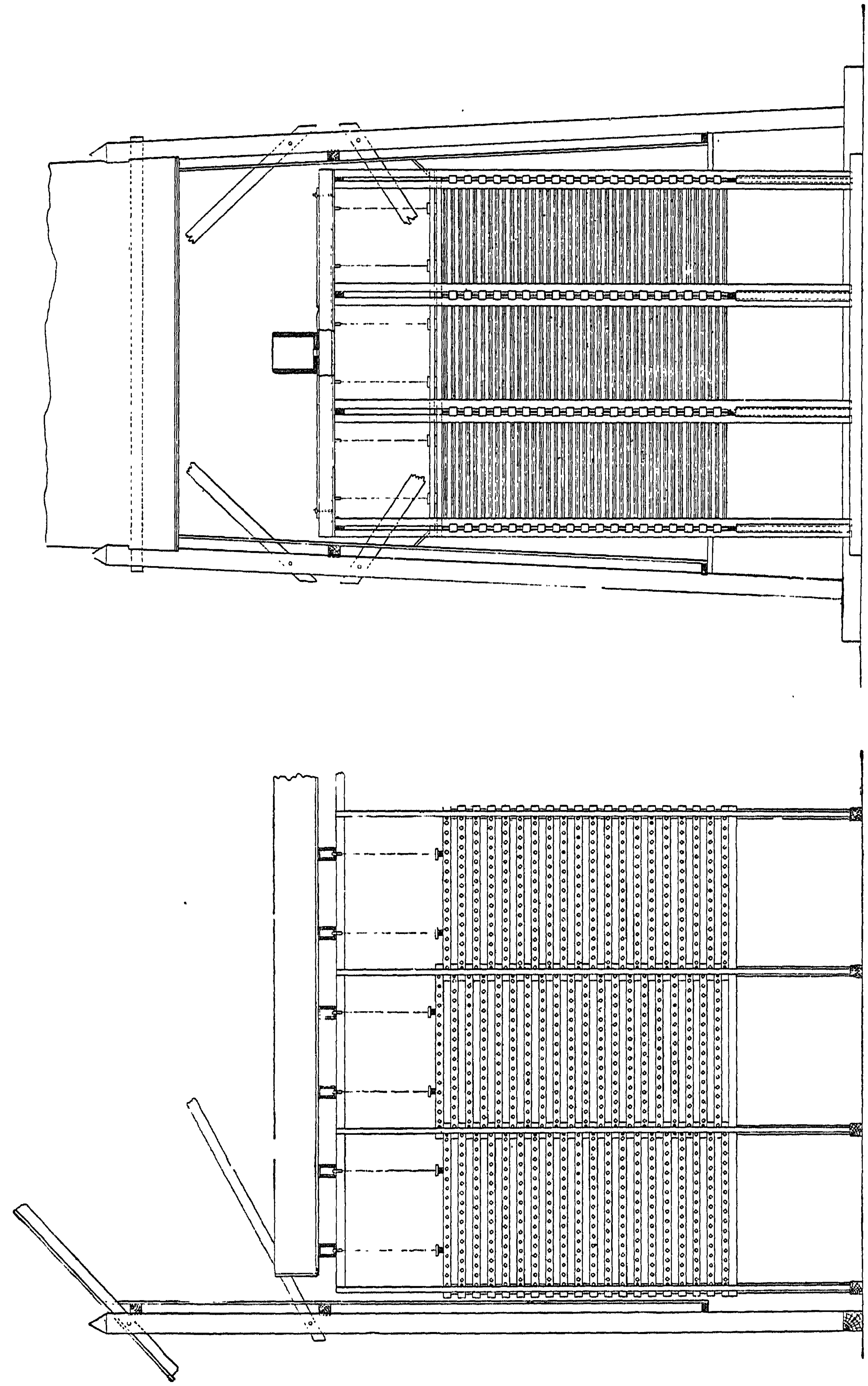


Fig. 33.—Arrangement of Splashbars in Cooling Tower by the Davenport Engineering Company

be induced to come into intimate contact with the water in the cooler, and the temperature and humidity of this air at inlet. Under average conditions a chimney cooler of 19 ft. × 19 ft. section, and about 60 ft. high, can deal with about 20,000 gall. of water per hour. A cooler 54 ft. × 26 ft. by 65 ft. high can cool about 100,000 gall. per hour, and to cool 300,000 gall. per hour requires about 164 ft. × 24 ft. by 70 to 80 ft. high.

The tank under the tower is usually built of concrete when erected at ground-level, and this also forms the foundations for the tower. But if the tower is elevated the tank may be built of up cast-iron sections bolted together. The capacity of the tank ought to allow the tower to be operated at full load for, say, twelve hours without addition of make-up supply when surface condensers are used. With jet condensers the water of condensation mixes with the condensing water and little make-up supply is necessary, but fresh feed water is required.

The amount of air required for cooling the water may be estimated by the methods explained on p. 252, taking it that the air leaves the water at the top at a temperature 10° to 15° F. below that of the entering water, and with relative humidity of about 85 per cent.

Except for the influence of winds the current of air through the tower is induced by the difference of density of the mixture of air and vapour in the tower and that of the outside air. If there were no resistance to the flow of the air the theoretical velocity of flow would be given by

$$v = \sqrt{2gH\left(\frac{\rho_0 - \rho_1}{\rho}\right)},$$

where v = velocity, feet per second,
g = acceleration due to gravity, 32 ft. per second²,
H = height of cooler above opening for inlet air, feet,
ρ_0 = density of external atmosphere,
ρ_1 = mean density of gases in tower,
$\rho = \frac{\rho_0 + \rho_1}{2}$.

To allow for frictional resistance to the flow of the air the above formula may be modified to

$$v = \sqrt{\frac{2gH}{(1 + F)}\left(\frac{\rho_0 - \rho_1}{\rho}\right)}.$$

The value of F may be taken to be about 18.

With low-level condensers the cooling tower is usually arranged with the base at about ground-level if the necessary area is available, but if a barometric condenser is used the tower may be placed in an elevated position. One arrangement, due to The Davenport Engineering Co., is shown in fig. 34. The tower has been placed over the boiler house at such a level that the injection water is raised and injected because of the vacuum in the con-

denser. When the tower is arranged at floor-level a pump would be necessary to inject the water into a barometric jet condenser.

When ground area is very restricted fan coolers may be adopted. The

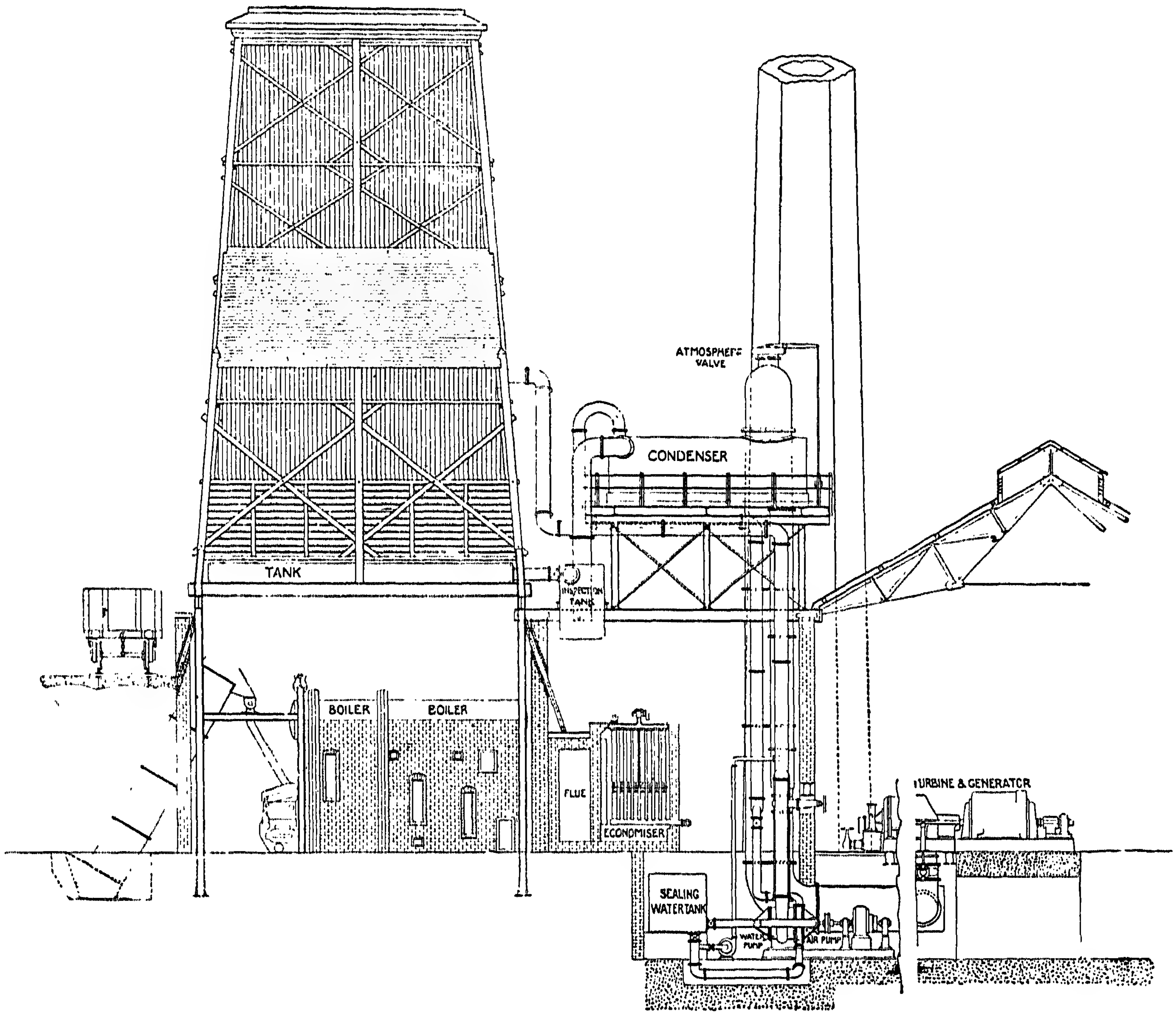


Fig. 34.—Arrangement of Cooling Tower and Barometric Jet Condenser

air is forced into the base of the tower by fans, and the increased current of air due to the fans increases the cooling capacity accordingly. The power required by the driving motors and the necessary attention and upkeep constitute a serious drawback, and this type of cooler has not found much favour except for the conditions mentioned.

THE OPERATION OF LAND POWER PLANTS

BY

JOHN W. JACKSON

M.I.Mech.E., A.M.I.E.E.

The Operation of Land Power Plants

Coal Crushing.—One of the most important problems before the power-station engineer is that of burning what is usually termed "low-grade fuel",* and a considerable amount of success has been met with in the working of this problem. A certain power station with which the writer is connected is now running on a very low grade of coal. Rather more than half of the total fuel supplied to the station consists of so-called "duff" coal. Duff is the finest of the dust coals, and 95 per cent of it passes through a $\frac{1}{4}$-in.-mesh sieve. The remainder of the coal comes direct to the power station as it is mined, and is known as "splint" or "coarse" coal. It is delivered in lumps about as large as a 7-in. cube; but sometimes the lumps run to a length of even 18 in. In order to secure the best combustion, it has proved desirable to crush this large coal to a size not greater than $\frac{1}{2}$-in. cube. Various experiments were made with coal crushers, and a successful type of crusher is that made by Edgar Allen, of Sheffield (fig. 1). It is of the rotary-roll type, the rolls being made of manganese steel. Three pairs of rolls are required to reduce the coal to a suitable size. The coal is then allowed to fall through a rotary coal filler on to a conveyor of the bucket type. The conveyor bucket then passes under another rotary filler from the duff bunker, and a quantity of duff coal is poured into each bucket. In this manner, a good mixture of the fuels is automatically conveyed to the boiler-house bunkers.

It may be asked why it is necessary to crush the splint coal so fine. There are several reasons. Although the splint may have calorific value of 10,000 B.Th.U. per pound, it has been found difficult to burn this fuel on ordinary travelling-grate mechanical stokers if it is larger than $\frac{1}{2}$-in. cube. With 2-in. cubes, the fires could not be kept alight for even ten minutes. The second point is this. Under ordinary circumstances, a piece of coal takes about half an hour to travel through a modern furnace. The whole of the carbon could not be burnt up in half an hour, unless the coal were finely crushed, while, if the carbon were rejected to the ash chute in

* Low-grade fuel has been defined as fuel which contains more than 25 per cent of ash and 10 per cent of moisture, or which passes through a $\frac{1}{4}$-in.-mesh sieve. *The Use of Low-grade and Waste Fuels for Power Generation*, Kershaw (Constable).

appreciable amounts, there would be the utmost danger of re-ignition, and of a blast furnace being created in the ash chute. The heat thus generated would bring the whole of the back end of the furnace down in a molten mass.

The Damping of Coal.—A certain amount of trouble was experienced

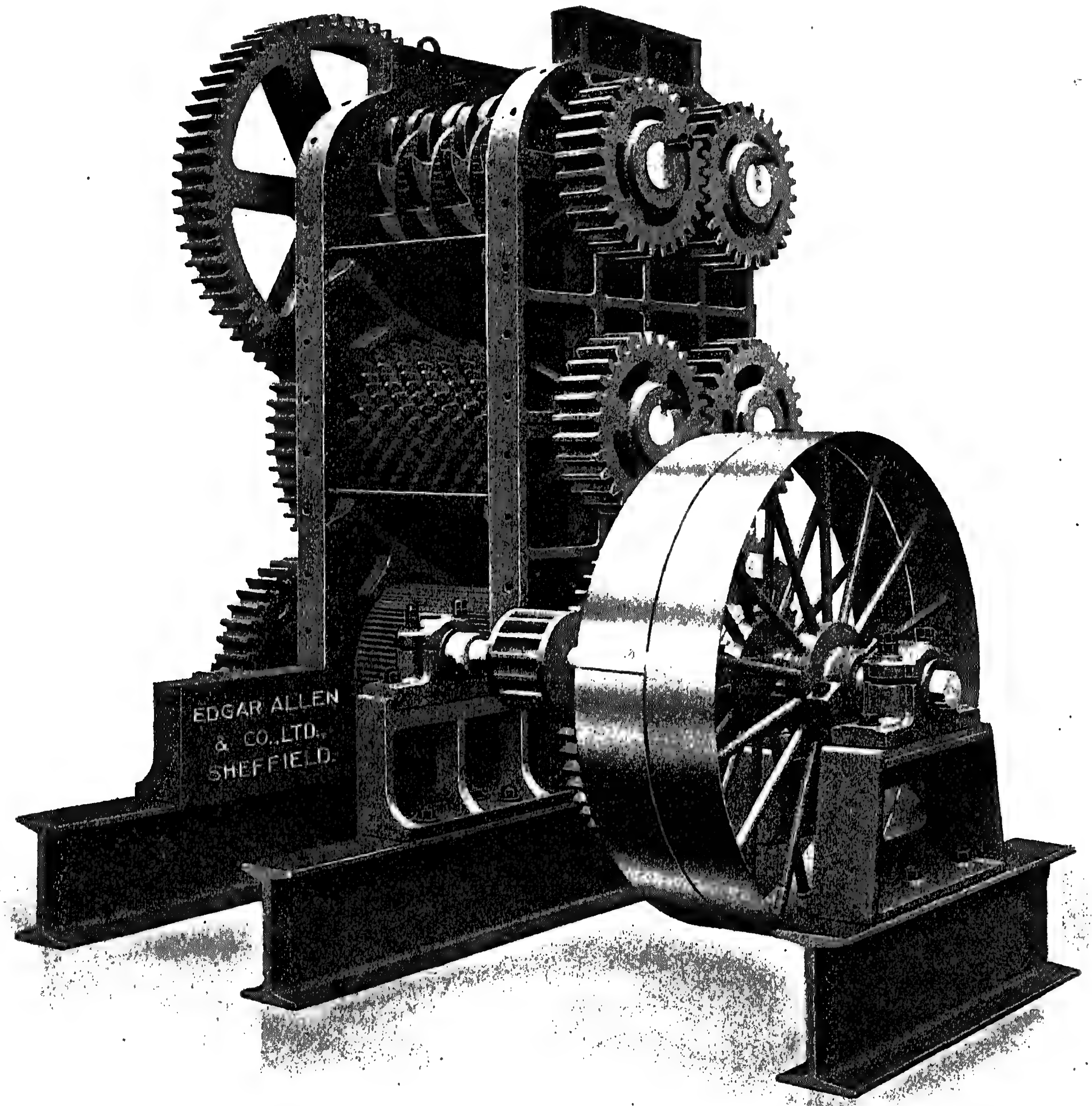


Fig. 1.—Coal Crusher

with the fuel prepared in the manner described, until a practical way was discovered of damping the whole of the fuel before it was put into the boiler-house bunkers. This damping, which adds perhaps 2 or 3 per cent of moisture to the coal, was found to be best carried out while the coal is on the conveyor, and before it is dropped into the bunkers. The action of dropping the coal and the water together from the conveyor into the bunker proved to give all the stirring that was necessary to secure a proper mixing.

The coal is led from the bottom of the boiler-house coal-bunker by means of chutes into travelling-grate stoker hoppers, whence it is fed automatically to the travelling grates, as required, according to the load.

Practically nothing but ash remains in the mass which passes off the grates into the ash hoppers. A good sample of ash will contain from 5 to 10 per cent of " combustible ", i.e. material capable of being burnt, and this is equivalent to a loss of from 1 to $2\frac{1}{2}$ per cent of the total coal fired.

Mechanical Stokers and Furnaces.—It frequently happens that the stoker gear, as designed, does not give the best service when dealing with special fuels.

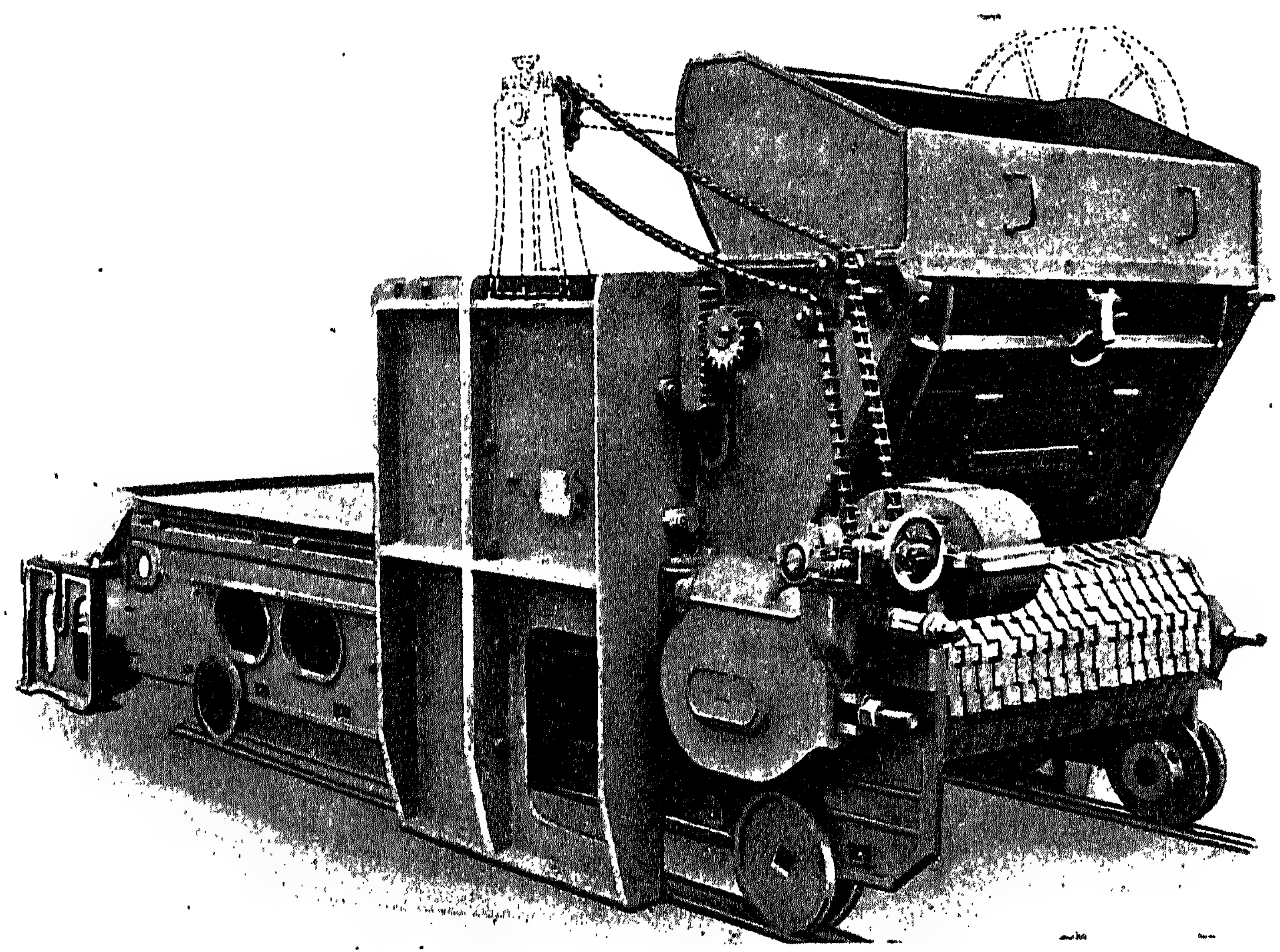


Fig. 2.—Babcock & Wilcox Patent Improved Mechanical Chain-grate Stoker

The importance of being able to deal with what we have described as low-grade fuel will be realized when it is said that many thousands of tons per annum used to be left in the abandoned portions of the coal-pit, and such of it as was brought " to bank " was, as a rule, dumped on to the pit heap. There was thus a large supply of cheap waste fuel available if it could be burnt.

Considerable advance has been made in the design of mechanical stokers. The moving-grate type of stoker has, however, proved to be by far the most popular. Many engineers are so convinced of its superiority over other types that they refuse to consider any other type of stoker. There are, however, considerable differences in the way in which the moving grate is operated. There is one moving grate which can be found in nearly every one of the big power stations throughout the country (fig. 2). This grate is used so universally because it deals with " the average fuel " successfully. One of the most successful types of moving grate is that using a particular design of link known as the " Parker link " (fig. 3). This link

is a considerable advance on the square-ended links previously used. The great advantage secured is that the amount of fuel which dropped through the grate, into the pits below it, was reduced to considerably less than half. The importance of this point will be realized when a station burning say 5000 tons of coal per week is considered. After the old square-ended link had been in service a little time, and had got worn, it was found that as much as 30 per cent, or almost 1500 tons of coal per week, fell through the grates into the pits below. All this coal had to be picked up again and returned to the stokers by hand labour. The Parker link, after an equal length of service, was found to reduce the amount of droppings to between 12 and 15 per cent, or say 600 tons of coal per week. An amount of even 600 tons per week is considerable, and the author set to work to secure, if possible, the advantages of both these links, and an improvement on them. A new link was designed, and after nine years' service the droppings in a grate fitted with it were found

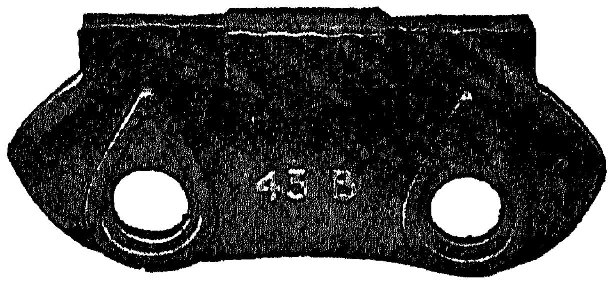


Fig. 3.—Parker Link

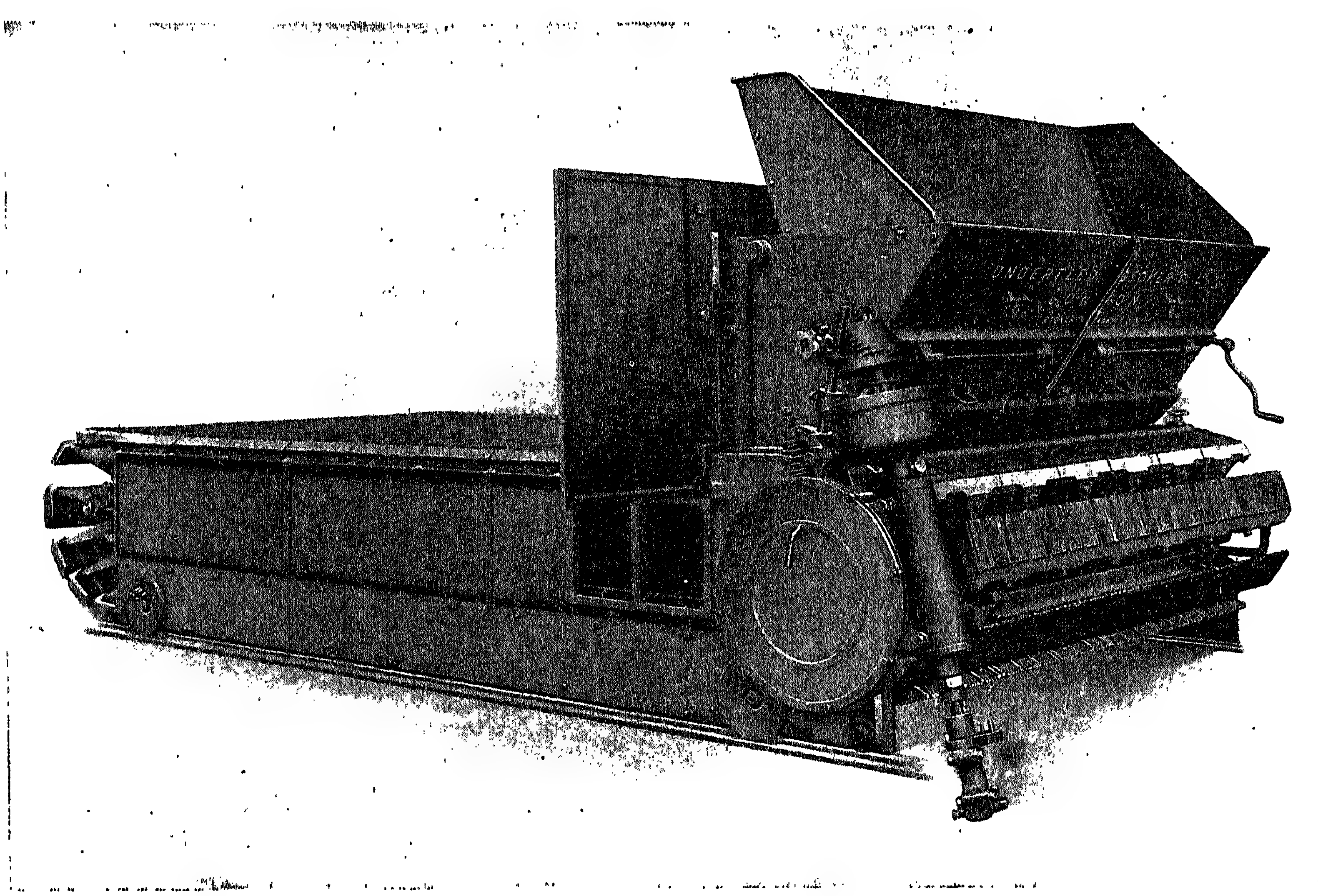


Fig. 4.—Type "A" Travelling Grate

to be 4 per cent. Further, by a very simple arrangement of the driving links, breakages of these links have been reduced to a minimum.

Now the above grates are only suitable for induced-draught conditions. When low-grade fuels have to be burnt, this type of travelling-grate stoker is of little use as, under ordinary induced-draught conditions, splint coal by itself will not burn, and can only be kept alight where an expert fireman is available, and when he gives the furnace his entire attention.

A few years ago another type of travelling-grate stoker came on the market, which possessed several new features. The important one was that it used a forced draught and introduced the draught on the "compartment method" (fig. 4).* This system is as follows: at the point where the fuel enters the furnace, the draught coming in contact with it is practically negligible, but by the time the fuel has travelled 18 in. into the furnace, the full pressure of the forced draught, amounting to perhaps $\frac{1}{2}$ in. to 1 in. water gauge, is brought to bear. This is continued for 3 or 4 ft., and from that point to the rear end of the furnace, which may have a total length of

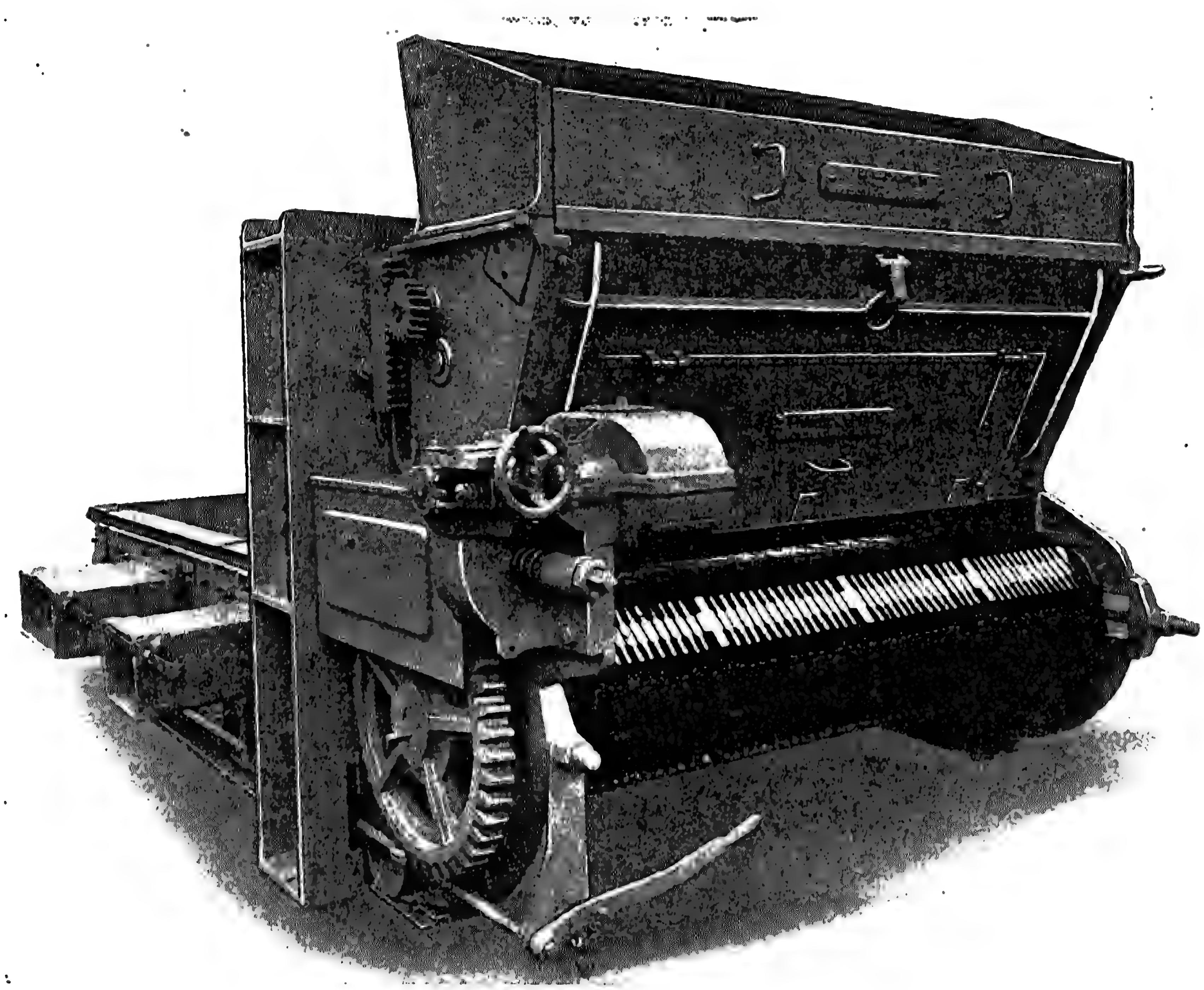


Fig. 5—Compartment-type Travelling Grate

14 ft., the forced draught is gradually cut off, so that for the last 2 ft. or so no air is passing through the grate. This arrangement gives the clinker time to cool down (before it is dumped down to the ash chute), although a violent stream of cold air is not flowing over it.

The boilers operating under these conditions are supplied with fans at the rear end of the boiler, which *induce* a draught through the boiler, as well as with fans which provide the forced draught. The net result is that the pressure in the furnace is kept at no more than $\frac{1}{10}$ in. below atmospheric pressure, i.e. $\frac{1}{10}$ in. water-gauge induction. This system is sometimes spoken of as the "balanced-draught system", the zero pressure (atmospheric pressure) occurring in the furnace chamber. It will be seen that

* This grate is made by the Underfeed Stoker Company, Limited.

the leakages of cold air into the furnace and flues are thus kept at a minimum, because the gas pressure inside is nearer to the atmospheric pressure than it would be if the induced-draught system were used. Further, it is possible to open the large inspection doors and examine the fire without upsetting the furnace conditions. On the pure induced-draught system, the moment the door is opened the state of gas flow is upset, the draught available on the fuel bed is reduced, and cold air pours into the furnace, and reduces the efficiency of the boiler to an alarming extent.

These remarks also apply to the ash tunnel where the doors have occasionally to be kept open, especially where a very low-grade fuel is employed, for comparatively long periods. From the point of view of efficiency of combustion, the balanced-draught system demands serious consideration.

Messrs. Babcock & Wilcox have recently introduced two types of forced-draught chain-grate stoker for burning low-grade fuel, one having a closed ashpit and the other with the compartments inside the chain. Further, angle-iron runners have been used for supporting the stoker links, with the result that the percentage of riddlings is said rarely to exceed some 5 per cent. Fig. 5 illustrates Messrs. Babcock's compartment-type forced-draught grate.

Ash Handling.—The question of how best to deal with ashes is a most important one.

When dealing with a low-grade fuel containing, say, 30 per cent of ash, the difficulty of burning the fuel so that the minimum of carbon is rejected with the ash is very considerably greater than it is with a fuel containing 25 per cent of ash. The increasing difficulty is out of all proportion to the increasing amounts of ash, and even the most expensive of ash-handling appliances demands the utmost consideration when low-grade fuels are to be burnt.

Perhaps the oldest ash conveyor was the iron-frame wheelbarrow, or bogie-wagon. Where this could be operated without great difficulty, it proved to be an extremely simple solution. Maintenance charges were very low, and it had many other advantages. But the size of the boilers increased, and it was found quite impossible to deal with ash on the firing-floor level, so that tunnels had to be formed underneath the back end of the boiler, where the ashes could be specially dealt with.

Ordinary shaker, push-plate, and tray-type conveyors were tried in turn. In the case of the shaker conveyor, the trouble proved to be that when a very hot flow of ash was deposited on it at any one point, it cockled up by distortion and bent itself into all kinds of shapes and so became unworkable. The troubles with the push-plate and the tray types of conveyor were very similar, and a large staff was necessary to keep them in repair. Dry ash is a very abrasive material, and causes rapid wear to all the bolts and fittings it rubs against. It was considered that conveyors of this type were doing extremely well if their average life was as much as twelve months.

With all the schemes so far discussed, dust and fumes were the great difficulty. A new system—the pneumatic system—did away with these

troubles. This system is worked by induced draught. An exhauster (fig. 6), somewhat similar to that used for producing the blast of blast furnaces, is used to suck the ashes along a pipe. This pipe passes under the ash hoppers. The ash passes from the ash hopper into a roll-type crusher (fig. 7), which breaks everything up to a size of about $1\frac{1}{2}$-in. cube, and thence it falls into an opening in the main ash pipe. This pipe is about 10 in. diameter, and the air is sucked along it at a speed of about 90 to 100 miles per hour. This draught quickly sucks the ash into the pipe,

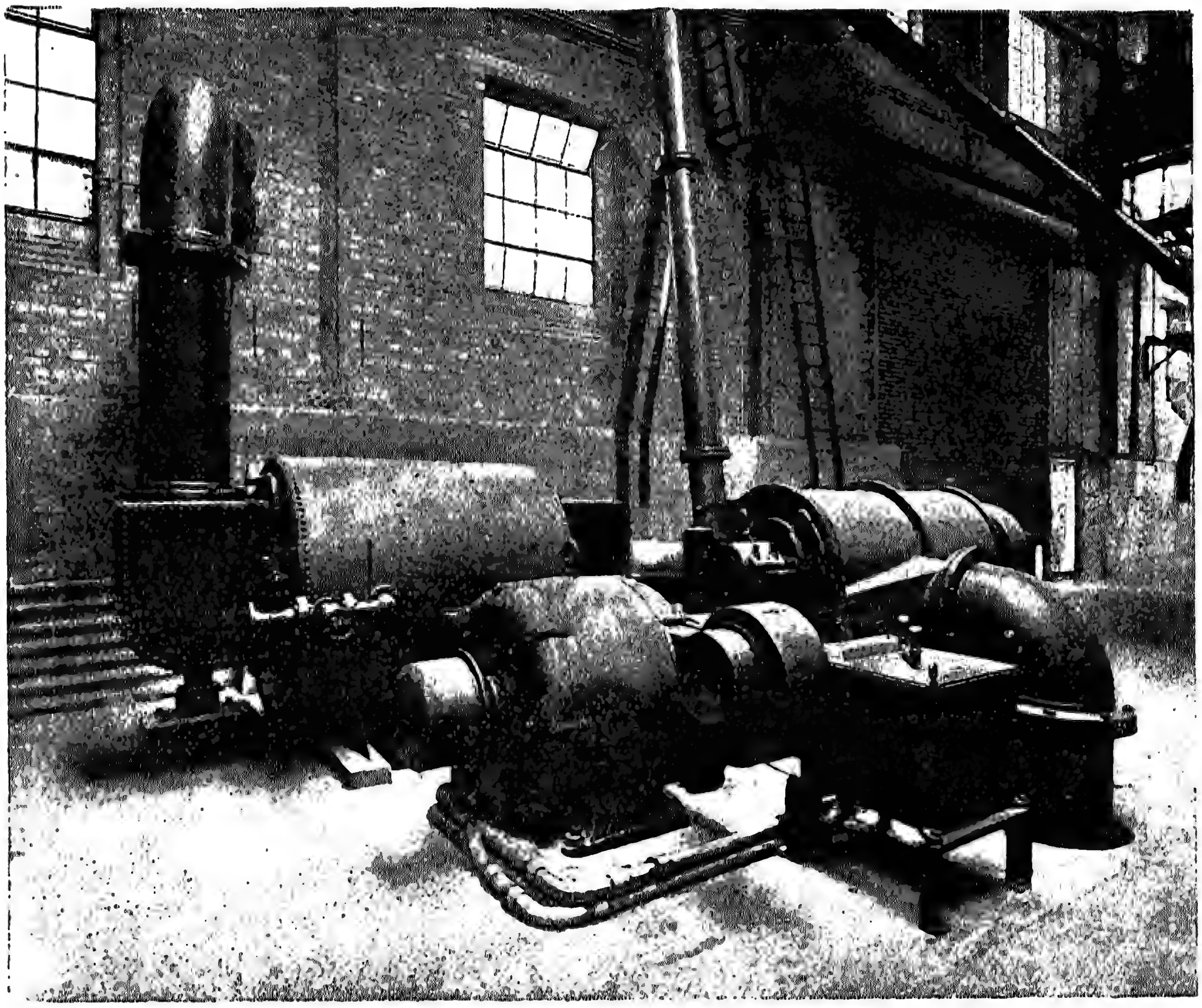


Fig. 6.—View showing Exhauster, Motor, and Dust Catcher with connecting Piping

along it, and up the vertical pipe to a height of even 40 or 50 ft., where it is deposited in the ash receiver (fig. 8). When the ash receiver is nearly full, the exhauster is stopped and a door is opened at the bottom of the ash receiver through which the ash falls into a railway wagon. It is usual for these receivers to be built for a capacity up to about 50 tons of ashes each. This system has entirely eliminated the evil of dust and fumes.

It will be noticed that the ash is never touched by man. The duty of the men is merely to start up the machinery and to guide it. A certain amount of expert attention is however necessary with this plant (fig. 9). This system has, up to the present, proved to be a good method, but expensive in upkeep and maintenance, for dealing with ashes.

The other simple method of using a standard push-plate conveyor inside a trough filled with water, as applied by the Underfeed Stoker

Company, has proved to be the most efficient means for dealing with large quantities of ash. where the plant is required to operate continuously, the attention during operation being low and the upkeep and maintenance charges much the lowest of any method yet tried. This system has, up to the present, proved to be by far the best method of dealing with ashes.

Boiler Cleaning.—The best way of reducing expense in the cleaning of boilers is to see that the water used is suitable or is made suitable.

Of the old Lancashire type of boiler very little need be said. The chief

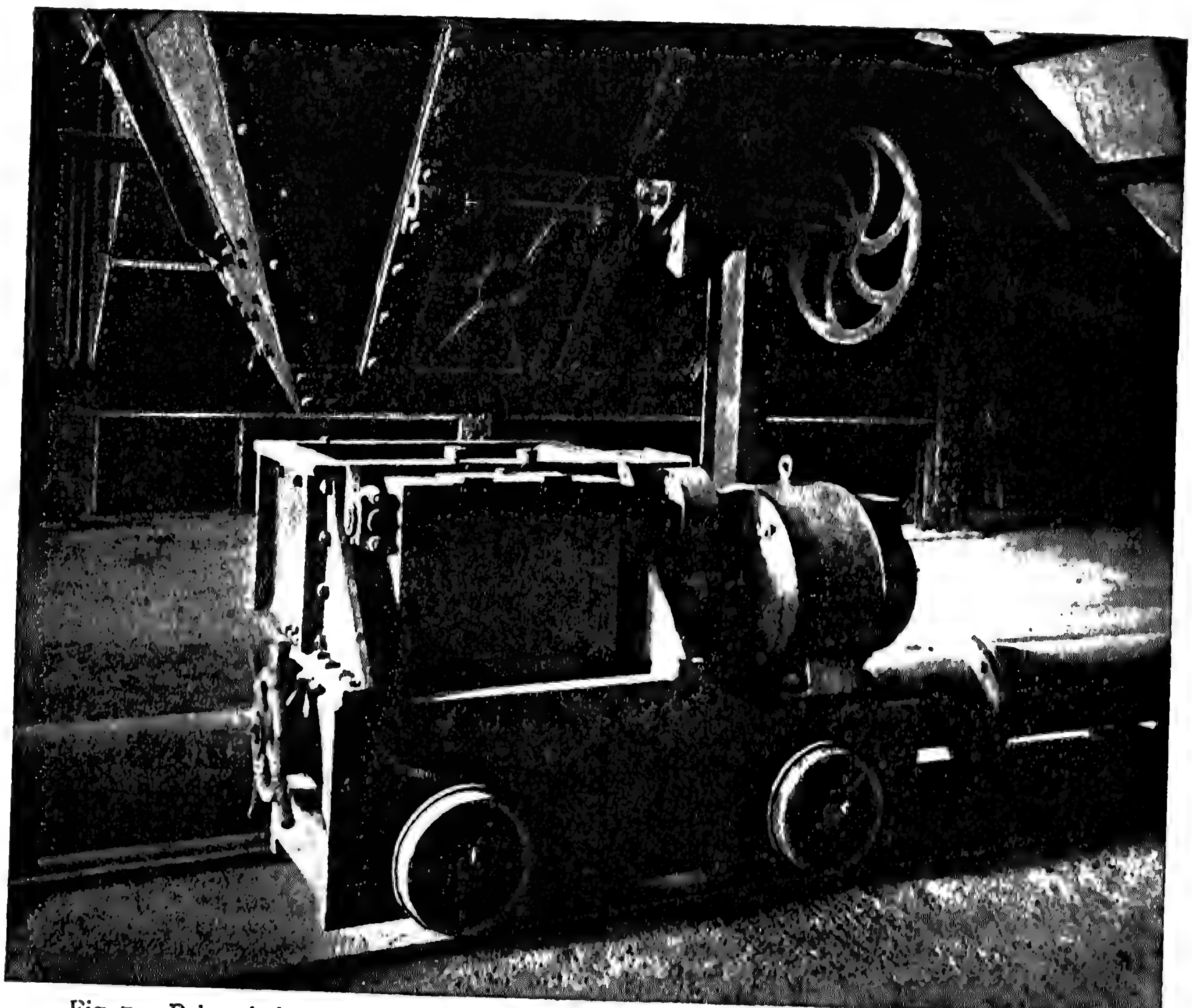


Fig. 7.—Babcock & Wilcox Patent Ash-conveying Plant. View in Ash Tunnel, showing Breaker and Ash Shoots

method adopted for cleaning these boilers is to chip the scale off the fire tubes with a scaling hammer, and finally to brush the tubes with a steel-wire brush. Since the fire tube expands and contracts more or less in service, the scale does not adhere very firmly, and so can be removed readily. Where the water-tube type of boiler is used, the expansion and contraction of the tube does not act on the scale so favourably. The scale is deposited inside the tubes, and as the boiler cools down, the scale is compressed into firmer contact with the metal of the tube.

It is physically impossible for any type of hand-scaling hammer to be used, and so a specially constructed cleaner has to be used. There are two systems in regular use. One is operated by compressed air. A pneumatically operated piston drives a number of tiny chip hammers or cones which are attached to the end of a hose-pipe and worked up or down the tube

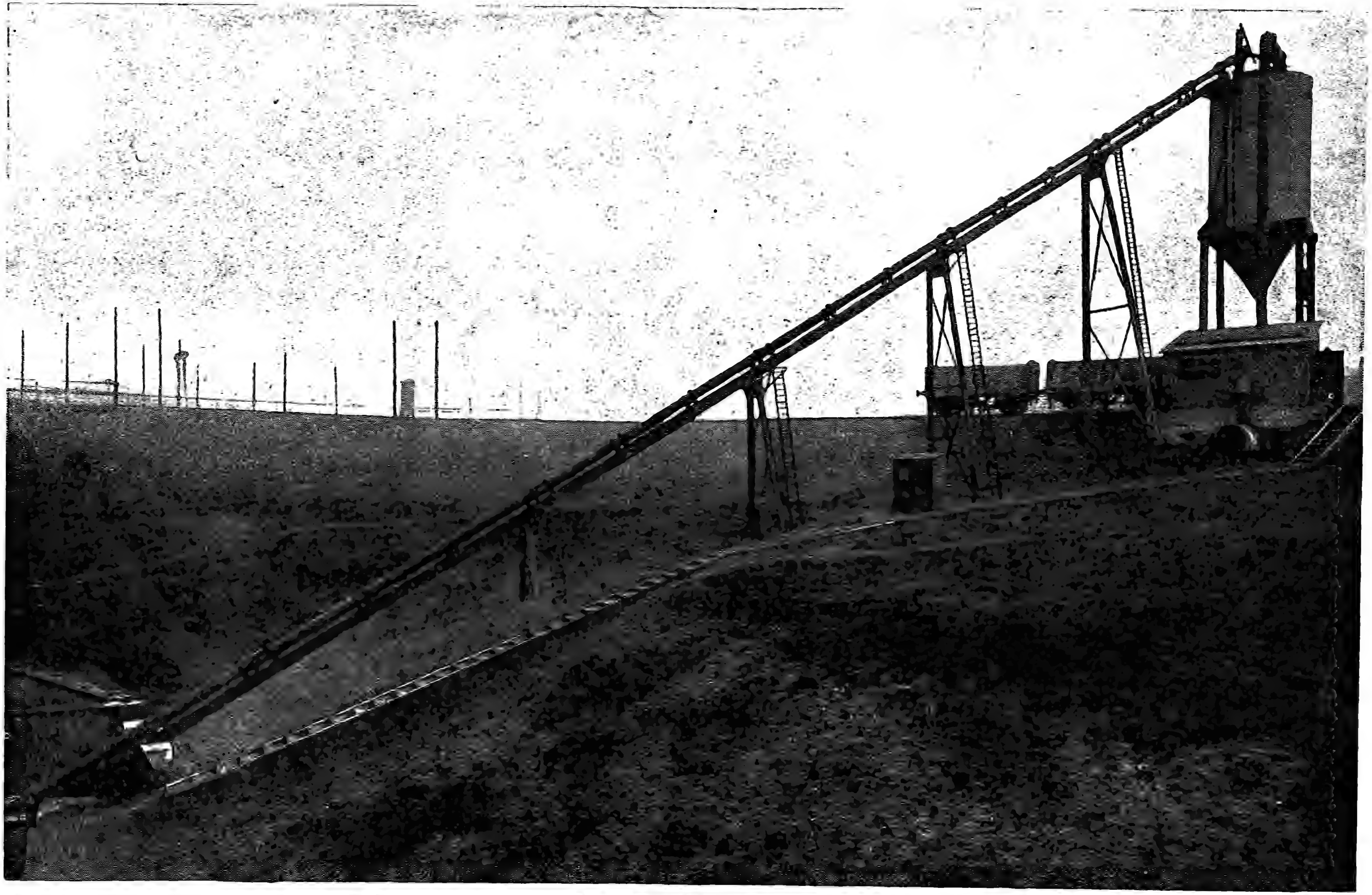


Fig. 8.—View of Ash Receiver and Connecting Piping of Plant handling 10 tons of ashes per hour

as desired. The exhaust air from the machine blows the dislodged scale away, and keeps the path clear for the cleaner.

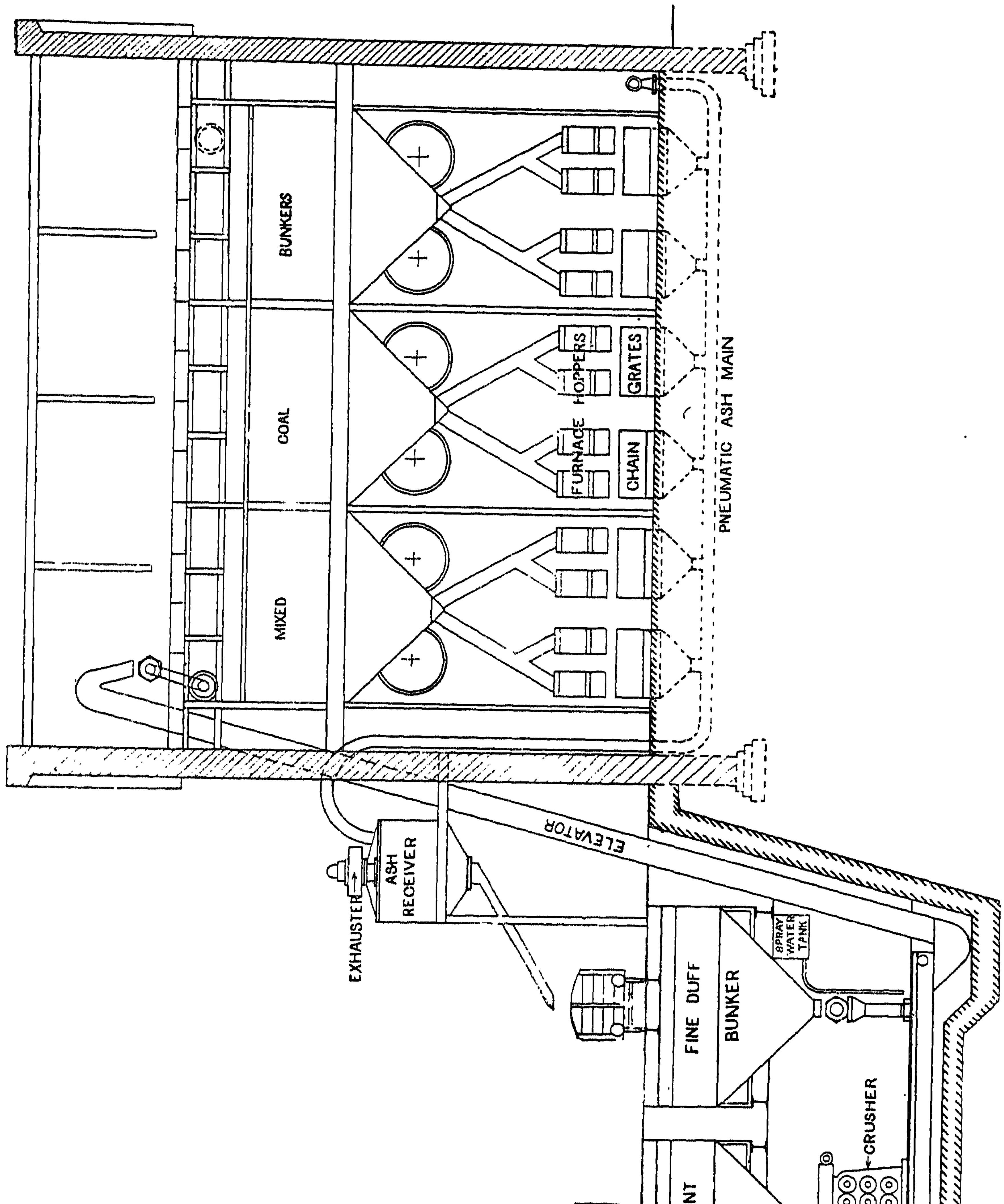


Fig. 9.—Ash- and Coal-conveying Installation

The other method is worked by high-pressure water. A small water turbine is used as the motor to drive the chip hammers or scale removers (figs. 10 and 11). Owing to the high cost of high-pressure water from town mains, it usually pays to install a special pump, or, failing that, to

adapt one of the regular boiler feed pumps so that it can be used on this work, when required, by isolating it for the time being from the remainder of the feed-water system. The best pressure for working the water turbine appears to be about 100 lb. per square inch. Here, also, the water is supplied to the turbine tube cleaner through a flexible hose-pipe, usually of the rubber armoured type. The turbine is passed down the boiler tube and can be controlled, as to position, as easily as the pneumatic type. The exhaust water washes the dislodged scale away, and this scale is caught in a water trough which is placed under the bottom headers of the boiler tubes.

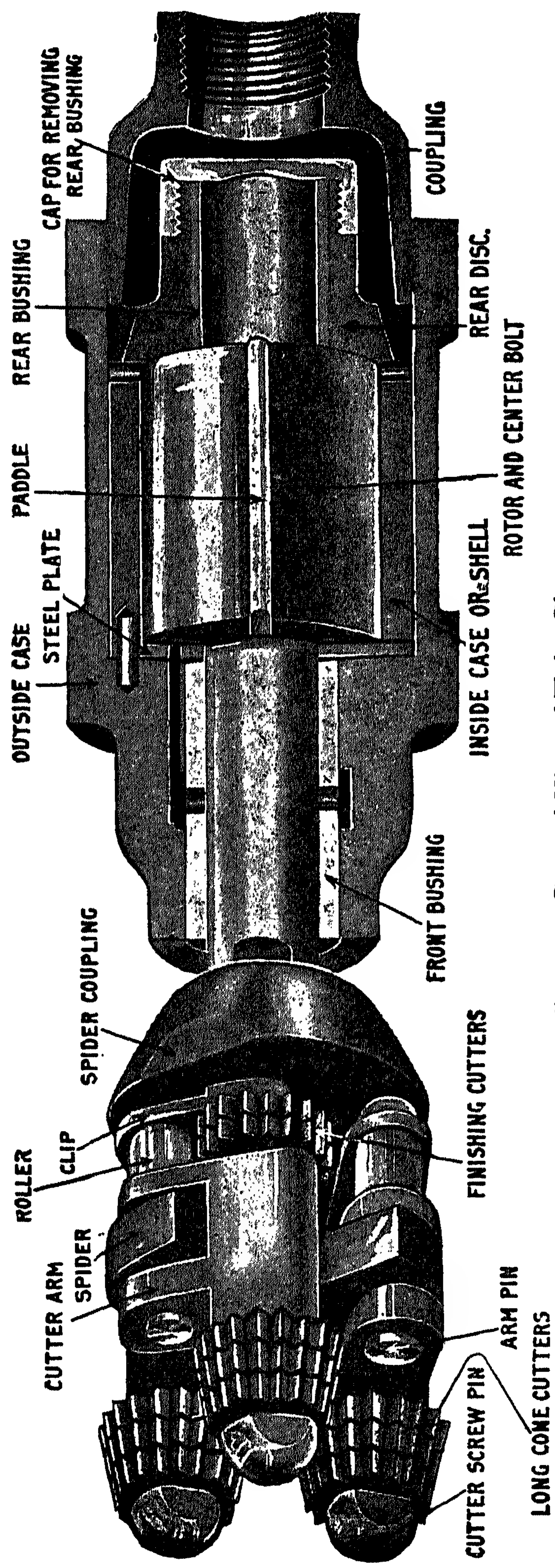


Fig. 10.—General View of Tube Cleaner

Economizers can be satisfactorily cleaned by using the scraper type of gear. This consists of a steel blade with a cutting edge at each end, mounted on the end of a long rod and kept apart by means of a spring, the spring keeping the cutting edges in contact with the boiler or economizer tube.

The scale can sometimes be removed by steel-wire brushes, mounted on long rods, if the scale is soft enough, but neither scrapers nor brushes are much use where the scale is hard and closely grained.

But " prevention is better than cure " and a hard scale need not be formed if proper boiler water is used.

There are many appliances and fluids

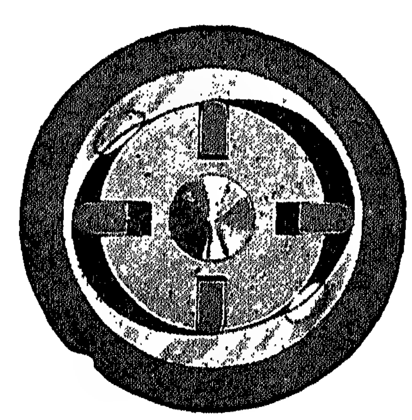


Fig. 11.—Cross Section of Boiler-tube Cleaner

on the market for the treatment of boiler feed-water. The claims made for some of these processes and fluids are wonderful and startling. Where a boiler composition is used with the water, the softening of the water naturally takes place in the boiler, and among other functions the boiler serves the turn of a water softener. Now most people will agree that the particular job a boiler is installed for is to raise steam, and not to soften water. It can become a positively dangerous piece of apparatus

when used in this manner, because the scum and mud liberated within the boiler may cause foaming, or priming, so that the superheater may be completely choked up. If the steam is not superheated, the foam is carried over to the engine or turbine, and may cause very bad water-hammer in the engine cylinders, and even blow the cylinder covers off altogether. With turbines, the blading may strip.

The author has, during the last twenty years, made a very special study of feed-water treatment. During this period many boiler compositions have been tried and discarded as being, in many cases, worse than useless where high-pressure steam is generated. It has been found that the old method of water softening by means of freshly burnt lime and carbonate of soda is the real scientific method. This method can be controlled with the utmost precision, by analysing the water frequently if it varies rapidly in composition, as it does when taken from an ordinary inland river. The analysis should be carried out about three or four times per day of twenty-four hours, and the analysis need give no more than the amount of total hardness in the water, and the amount of temporary hardness. Having determined these quantities, an alteration is made to the filter plant, as required, so as to increase or decrease, separately or together, the amount of lime or soda introduced per thousand gallons of water treated. In this way (1) no more of either of these two chemicals is put into the water than is absolutely necessary, (2) the bulk of the deposit is left in the filter plant instead of in the boiler, and (3) the water is neutralized so that it is nearly soft, but not quite, the outstanding hardness being compensated for by the small amount of soda that is put into the boiler in excess. It has been proved that boilers can be left on load for twelve months at a time without (1) requiring to have the turbine tube cleaner put through the tubes at all, (2) without losing any boiler tubes from the formation of scale, or internal corrosion, and (3) without having trouble with the brass mountings of the boiler, such as the water gauges, &c. The boilers, during the whole of their normal working period, can be kept at their normal full-load rate of evaporation. The essential points of the method are (1) frequent examination of the water, and (2) alteration of the amount of chemicals added to the water as and when required.

The chemical methods by which the necessary analyses may be readily made are described in detail in this volume (ENGINEERING CHEMISTRY).

Valves and Boiler Mountings.—The question of keeping the valves and boiler mountings tight is one that requires considerable thought and attention, and the first policy to adopt, where low maintenance costs are especially desired, is to use the highest class of fitting only. There are, unfortunately, many steam valves on the market which are made of material that has not been treated on recognized scientific lines either as regards the nature of the material used for constituent parts or the heat treatment it receives. Failure in either of these directions is fatal in high-pressure work. It has, in fact, proved necessary where highly superheated steam is used, to use alloys that contain no yellow metal whatever. These alloys contain

high percentages of nickel, and, some of them, a certain amount of platinum and chromium.

Similar remarks are equally true of the valves used on the water side of the boilers. If more than sufficient soda is used the brass water-gauge and other fittings may be attacked, and so become expensive to maintain. The maintenance of valves becomes a very serious matter where inferior material is used, and the cost of maintenance per annum may quite easily prove to be considerably more than the original capital cost of the valve or fitting involved.

Priming of Boilers.—The priming of steam boilers may become serious if an excessive amount of soda is present in the boiler water. Some waters are naturally high in soda. The only step to take in that case is to blow out the entire water contents of the boiler to the drain frequently. This operation is known as "blowing down". In other cases this excess may be due to too much soda having been added for water-treatment purposes. It may, however, have been added in the form of "Somebody's" patent boiler composition. If the power station depends upon a tidal water or sea water for its circulating water system, leaky condenser tubes may allow sufficient salt to enter the condensed steam to cause priming. It will be remembered that the condensed water is returned direct to the boilers by the boiler feed-pumps in a condensing system—and this system is used for all modern plants. Contamination with salt water has proved dangerous in many cases, and where the turbines have had their relay valves operated by steam, the governors have sometimes been rendered inoperative, and have allowed turbines to run away until the emergency governor has come into operation. In some cases, this trouble has led even to the destruction of the turbine plant.

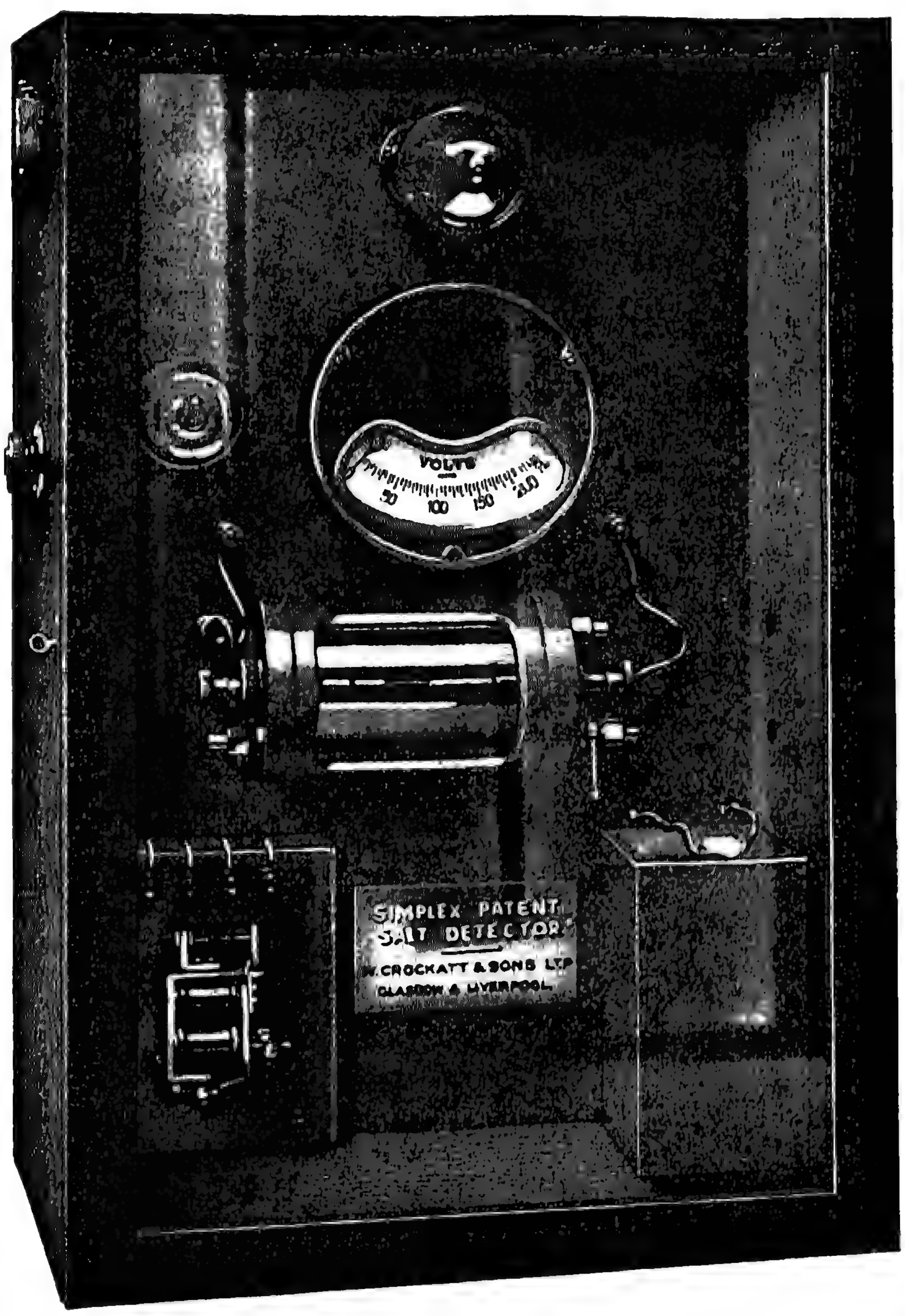


Fig. 12.—Feed-water Density Indicator

There are many excellent devices on the market for indicating the condition of the condensate. Some of these devices test the water continuously and cause an alarm bell to ring directly the water reaches a predetermined value in density (fig. 12).

Troubles with Turbines.—As has been previously pointed out, turbines may run away because the governors are prevented from acting by the effect of impure boiler feed-water. The kind of accident that may arise with governor gear is well illustrated by a trouble which actually arose with a turbine-driven boiler feed-pump. This little set consisted of a centrifugal pump direct-coupled to a single-wheel impulse turbine. The pump was delivering water at a higher pressure than that of the steam which was used to supply the turbine. The governor gear was operated on the "plunger principle", i.e. a piston in a cylinder was controlled by steam on one side and water on the other, and a spring was provided so that when the water pressure exceeded or fell short of a certain amount, the piston was driven upwards or downwards, and so controlled the admission of steam to the turbine. After a few weeks' work the governor plunger was not perfectly tight, and as the water on the one side was at a higher pressure than the steam on the other, the water leaked past it into the steam side until it overflowed into the main steam-pipe. As the steam was superheated, the water was quickly evaporated in the steam chest of the turbine pump, and as the water was not pure a deposit was left behind which settled on the emergency main governor valve gear. The pump had to be frequently taken apart and cleaned, but on one occasion it had fouled quicker than usual, and the main and emergency governor gear fouled simultaneously, with the result that the turbine ran away. The moral of this fact is—quite apart from questions of boiler corrosion—keep your boiler water absolutely pure. It is dangerous not to do so.

Naval engineers know the necessity of pure boiler water even more than power-station engineers. With the very fast-steaming non-superheating boilers which are used on destroyers, for instance, a little salt water in the boiler water may cause priming to take place, and put the destroyer completely out of action by damaging its turbines just at the critical moment when it requires its utmost speed.

Other troubles may overtake the turbine, such as the stripping of the blading in the case of the reaction turbine, or the fracture of the diaphragm nozzles, or even the wheels, in the case of the impulse turbine. These troubles may arise from sudden changes in temperature. The stripping of a turbine may also be caused by allowing the boiler water to get very muddy. Considerable quantities of mud are blown over with the steam through the superheater until it deposits on the blades. Cases have been known where the turbine has been completely choked with mud, so that the pressure put on to the blading has caused it to bend until it has fouled the standing blades, thus causing a strip. Or again, the failure of a reaction-type turbine may be due to the manner in which it is started up. Much care is necessary in this operation to see that the turbine is, as far as possible, equally heated up throughout its high-pressure portion. The most fatal thing is to warm it up comparatively slowly. This allows a little trickle of steam to flow through it at one particular point, say at the extreme top or bottom. This may be sufficient to heat the spindle up in that position, and to cause it to

expand more there than on the opposite side. This in turn causes the spindle to bend and the spindle is thus thrown sufficiently out of truth to cause the running blades to foul the fixed ones. The only safe way to heat up a turbine is to pass comparatively big quantities of steam through it and then to shut the steam off for a short period, thereby ensuring a uniform distribution of steam throughout the blading spaces. These gusts of steam can be repeated say once every half-minute for ten or fifteen minutes, by which time the turbine can be started up.

The impulse turbine is somewhat easier to start up than the reaction turbine. The impulse turbine does not require so long a time to get warm, and can be started up direct from cold without danger. The reason for this is that the hot steam is never allowed to come into contact with the big masses of metal as is the case with a reaction turbine, so that it is quite a common thing for an impulse turbine to be run straight up on load without any previous heating up. Should the turbine be designed so that it runs through a first critical shaft speed, and its normal speed is say 3000 r.p.m., its first critical speed will probably be somewhere round about 2000 to 2400 r.p.m. The proper procedure is to give the turbine sufficient steam to start it moving, and to allow it to run up to a speed of approximately 1500 r.p.m. quite slowly. When that speed has been reached and the operator is quite satisfied that the machine is ready to go on load, he should then open the stop valve freely, allowing the machine to run up to its normal speed as quickly as possible, thereby making sure that the turbine is not allowed to dwell on the first critical speed under any circumstances whatever. If the turbine rotor is allowed to revolve at its critical speed for any length of time, any little external vibration that happens to exist and to synchronize with the critical frequency may cause the spindle to run out of truth. This slight bend in the spindle will develop more and more at every rotation until it is sufficient either to strip the blading off the wheels or even to burst the casing. In its mildest form, this trouble may lead to a permanent bending of the shaft.

In shutting down a machine of this type there is only one sound way, and that is to shut off the steam straight off, and, as a rule, the machine will slow down fast enough to avoid any difficulty from its running through its critical speed.

It might be gathered from the above remarks that the impulse type of turbine is the better machine for power production. This is not necessarily the case. Other factors have to be taken into account in deciding which is the better type for any particular purpose.

It has been found that much less risk and expense is incurred by keeping a machine on load than by shutting it down and starting it up again if it is likely to be off load only for two or three hours at a time. The bulk of the troubles from which steam turbines suffer, develop during the period of starting up. This is due to the difficulty of obtaining uniform expansion of the masses of metal in the machine. Even when the greatest care is exercised, it is almost impossible to admit the steam in the same manner

as it is admitted when running on load, and it is better to run through the dinner hour and similar periods rather than to shut down the machines to save standby coal.

Lubrication of Turbines.—The lubrication of the modern steam turbine is almost entirely automatic.

The oil is circulated by means of a pump which is driven direct from the main shaft (fig. 13). This pump is usually of the rotary type, and often consists of two gear wheels. These pumps, simple and strong though they appear, are by no means free from trouble, and unless the oil can readily escape from each individual tooth as the corresponding tooth of the other wheel engages with it, serious vibration may be set up on the pump shaft, causing the key-ways to strip, or the pump shaft to break.

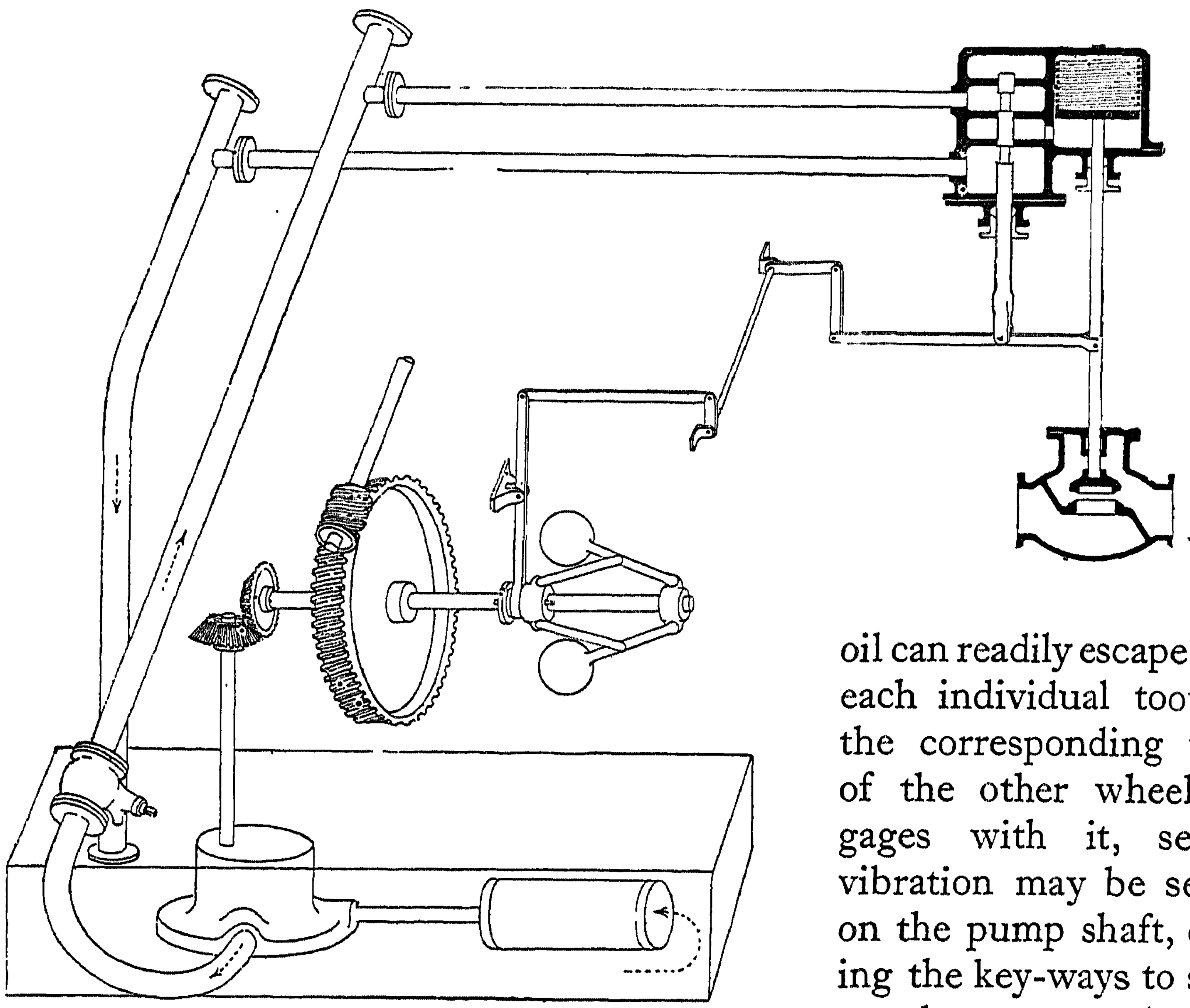


Fig. 13.—Diagrammatic Arrangement of Turbine Governor on relay system

There are many modifications of this rotary pump, but none, perhaps, are quite so simple as the one above described, and probably none are so free from trouble. An auxiliary oil-pump is usually fitted, which is directly driven by steam. It is generally used when the turbine is started up and shut down, but it can be used for supplying the turbine bearings with oil under service conditions, in emergency. It may be mentioned that failure of the system of lubrication is serious, since the bearing liners may melt and consequently the turbine spindle may drop sufficiently to foul the fixed blading.

Governors.—The emergency governor is usually mounted directly on the end of the main turbine shaft (fig. 14). It consists of a ring, the centre of which has been bored out eccentrically, and the difference in the weight of the opposite sides of the ring is compensated for by means of a spring.

When a certain predetermined speed is reached, the centrifugal force

arising from the unbalanced mass of the ring overcomes the spring and allows the ring to strike a trigger which releases the main-valve operating gear.

Probably the best type of main governor is that where the main valve is controlled by oil (fig. 13). This system by oil control lends itself very well to the requirements of the emergency governors set to shut down the main plant when the speed exceeds about 12 per cent above the normal.

Condensers.—The testing of condensers, especially where sea water is used as circulating water, is important. Testing for and replacing faulty tubes may cost as much per annum as the whole of the electrical plant.

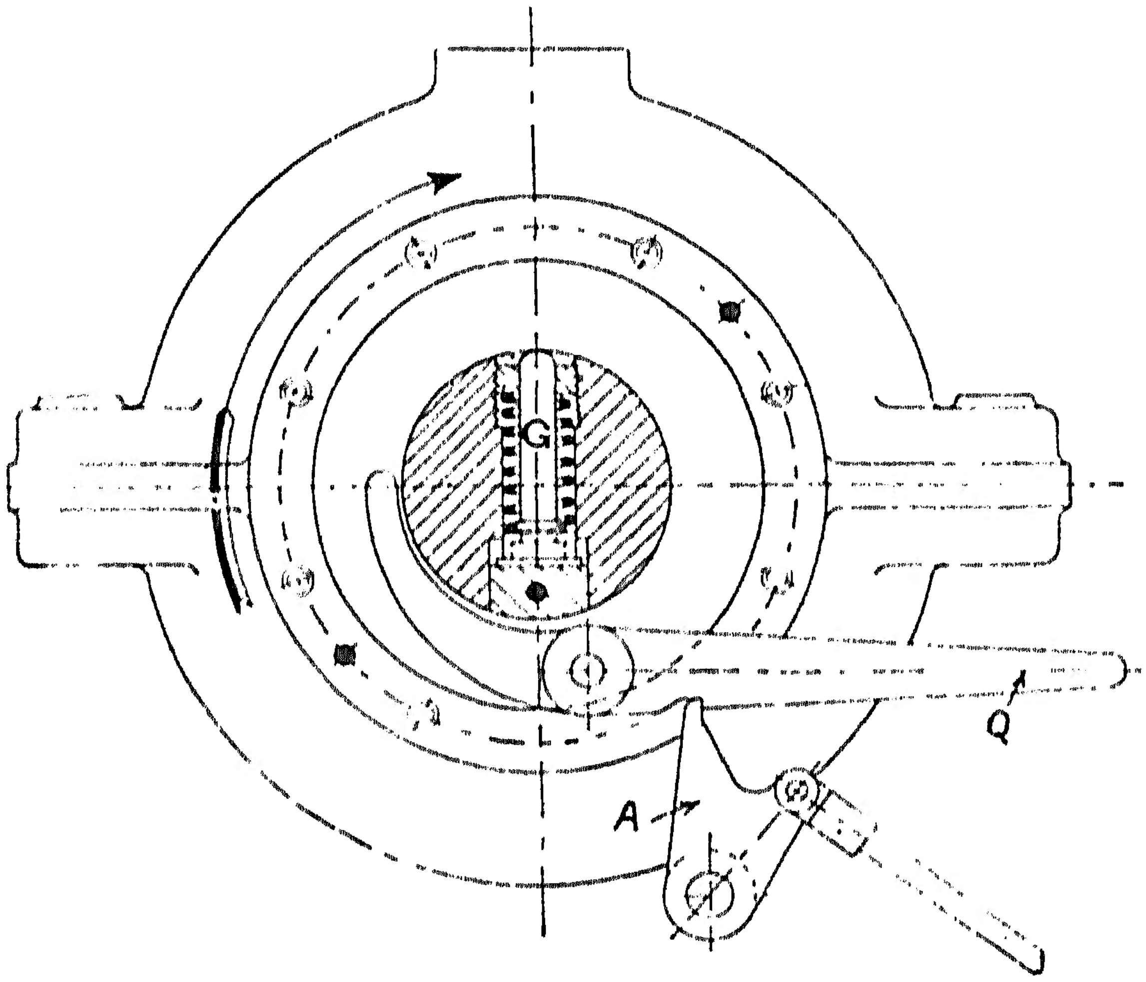


Fig 14.—Emergency Governor

The testing of condensers is important because of the havoc that may be wrought in the plant if salt water is permitted to get into the boiler water. It has been found that, should a leak develop which will cause a salinity of more than about 12 grains of salt per gallon, the water should be turned to waste until such time as the condenser can be tested and the leak stopped.

There are two simple methods of testing the condensate for salt when the plant is on load. One of these consists of drawing a sample of water and analysing it in the usual manner for salinity by the chemical process (Vol. IV). The other method is an electrical one. An apparatus can be arranged to give an alarm directly the salinity reaches a predetermined value. The apparatus is simple, and has been at work for some time, and up to the present it has given very good results (fig. 12).

When it comes to finding the leak in the condenser, the work usually has to be carried out during the periods of light load, when the turbine plant can be shut down. The air-pump connections are either blanked off, or a valve is provided that can be shut. The manhole is then opened at the top of the condenser, which allows the condenser body to be filled with fresh water. The manhole doors in the condenser end plates are also opened and an inspection is made of all the nipples in the condenser tube plates until the leak is discovered. It is quite a common thing to find that a number of tubes begin to fail simultaneously, and though each allows only a minute leak, together they are dangerous.

On a big condensing plant this test may take as long as six hours. It is by far the quickest method, however, that has yet been tried, and is the most certain in its results.

When a leaky tube is discovered, the ends are plugged and the tube placed out of action for the time being until enough tubes have been plugged to affect seriously the efficiency of the plant as a whole. By this time a new set of condenser tubes should be ready to go in. This is a very expensive method, but appears to be the cheapest one discovered. Many attempts have been made to repair condenser tubes by soldering up the holes, by brazing them, and by electric welding, &c. Up to the present all have proved useless. The difficulty is that of closing the hole with metal of exactly the same thickness as that of which the body of the tube is made. The metal must not project externally, otherwise the tube could not be fitted into its position through the tube-plates; on the other hand, if it projects internally, the usual cleaning brushes will not readily pass along the tube.

Air-pumps.—The "best" air-pump is apparently still to be produced. The reciprocating pump is becoming unpopular for several reasons. Firstly, the cost of its maintenance is high; secondly, its efficiency as a vacuum pump is low; and thirdly, it is unable to deal with large quantities of air, should a serious leak develop in the condenser system. Among its advantages is the low power it takes to drive it.

The "kinetic" pump is a good one. This pump consists of three centrifugal pumps mounted on one common spindle, working in conjunction with a steam jet. It may be run at speeds of 1500 to 2000 r.p.m. It usually has three simple bearings which require attention, but no more attention than is given to the ordinary ring oil bearings of an induction motor. The cost of maintenance is reduced to an almost negligible amount. Plants have been at work now for many years. Some plants, in particular, have been opened out once a year for examination over a period of three years, and have been put together again without requiring attention in any detail whatever, although running on an average of fourteen hours per day throughout the year. This kind of pump appears to have nearly reached perfection from the standpoint of low maintenance cost and reliability.

There are many other types of rotary air-pump on the market, such as the Leblanc pump. This pump is quite different from the "kinetic" pump, although it is a rotary pump. Its essential feature is a revolving wheel something like a turbine wheel. The blades pass a fixed jet of water and cut off slices of water which are thrown down the throat tube of the pump. The space between these slices is filled with air which is expelled by the momentum of the water imprisoning it.

When a condenser can be kept reasonably free from air leaks, and the boiler plant is also normally operated, i.e. where no excessive quantity of air is permitted to get into the steam system, this pump will give very good service, and maintain possibly quite as high vacuum as the kinetic pump; but it has the very serious disadvantage, that when a heavy air leak develops, the pump is unable to deal with the extra air and the vacuum falls off very severely, and may shut down the condensing system.

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